COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

September 1958

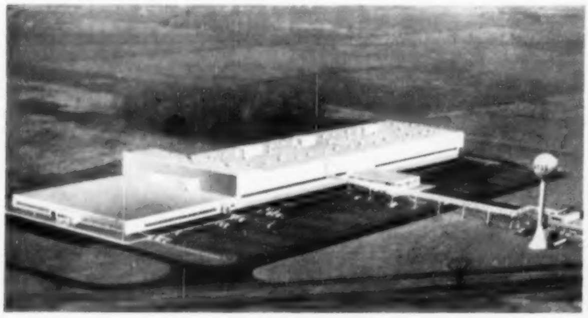
World's largest packaged boiler enroute to the Scovill Mfg. Co. power plant in the background. (See page 38.)

Largest Packaged Boiler

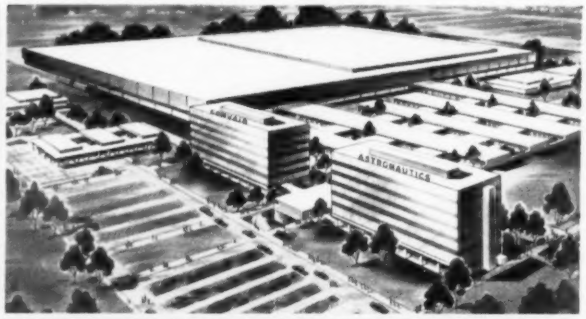
Paris Conference on Combustion

An Experimental Air Condenser

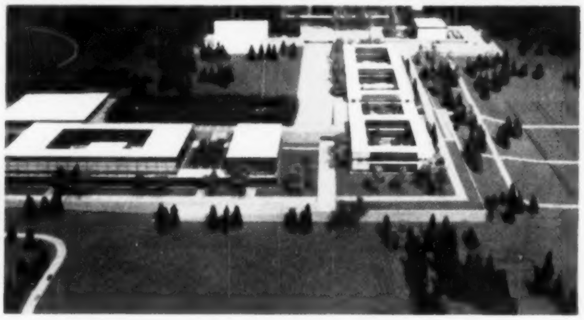
THE CROSS COMPANY—Fraser, Michigan
This automation equipment producer's new plant has two 12-million btu C-E Hot Water Boilers.



CONVAIR-ASTRONAUTICS—San Diego, California
New Convair space-flight research center has two 30-million btu C-E Hot Water Boilers.



RIVERVIEW COMMUNITY HIGH SCHOOL—Riverview, Mich.
Two 10-million btu C-E Hot Water Boilers heat Riverview's recently expanded high school.



U.S. AIR FORCE ACADEMY—Colorado Springs, Colorado
Choice of five C-E units here typifies widespread Air Force acceptance of high temperature hot water.

HIGH TEMPERATURE WATER

today's new idea in large space heating

In such diverse applications as industrials, large educational institutions, and military bases, high temperature water is finding rapidly increasing acceptance as an ideal means of heating large areas.

A big factor in bringing this trend about is The C-E La Mont Controlled Circulation Hot Water Boiler. Using the same principle as that applied by C-E in many of the country's largest utility boilers, this new boiler provides a degree of temperature control that makes it the most attractive method of heating in many cases. With a wide range of capacities—from 10 to 300 million Btu's—these boilers operate at water pressures up to 500 psi and temperatures to 470F, or higher. A C-E Hot Water Boiler can save from 10 to 20 per cent in maintenance and operating costs.

Combustion Hot Water Boilers in the smaller capacity range are completely assembled in the shop while the intermediate and large units are shipped in varying stages of assembly. This C-E practice greatly reduces erection costs.

So, if you are in the market for boilers, either for space heating or process requirements, it may prove greatly to your advantage to investigate the use of high temperature water as your heat source. Because individual needs vary, both steam and hot water have their place. Our engineers will be pleased to discuss either method with you or your consultants—impartially and with no obligation.

For further details on high temperature water boilers by C-E write for our catalog HCC-2.

COMBUSTION ENGINEERING

C-E

Combustion Engineering Building, 200 Madison Avenue, New York 16, N.Y.
CANADA: COMBUSTION ENGINEERING-SUPERHEATER LTD.

C-120-A

ALL TYPES OF STEAM GENERATING, FUEL BURNING AND RELATED EQUIPMENT; NUCLEAR REACTORS; PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 30 **No. 3**

September 1958

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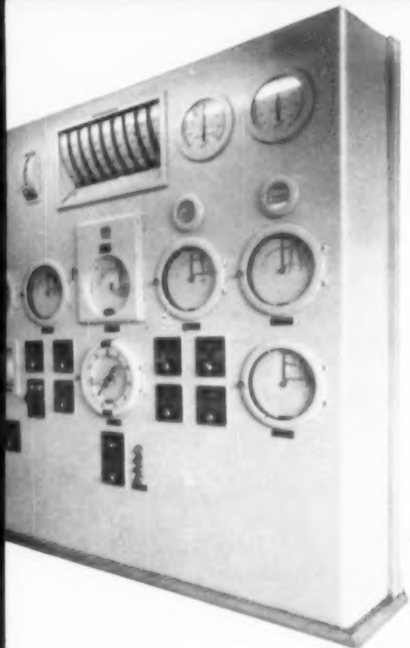
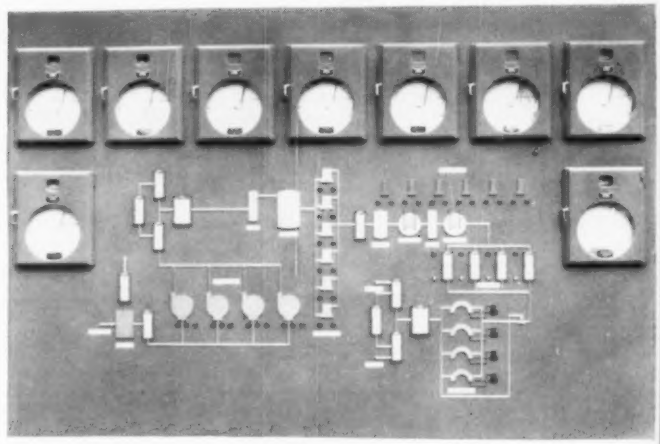
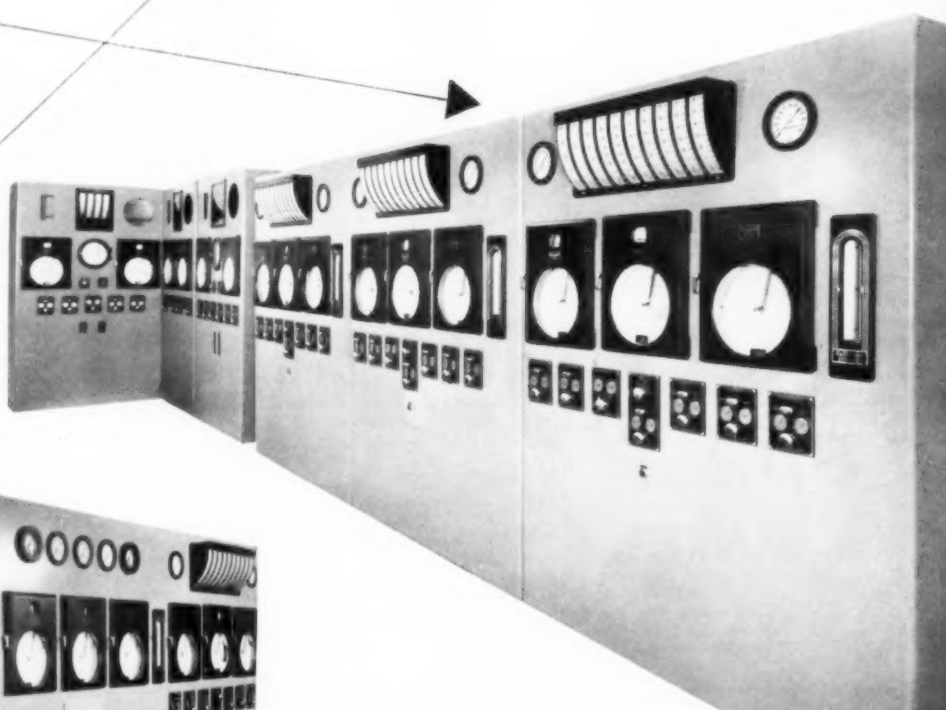
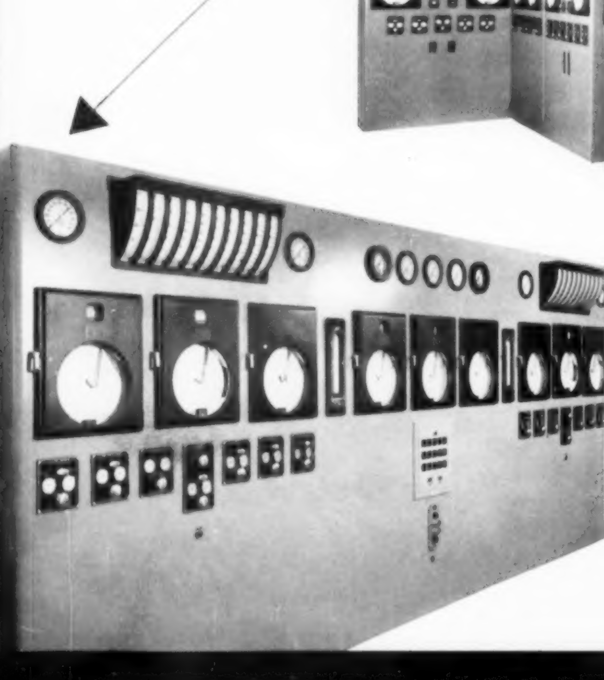
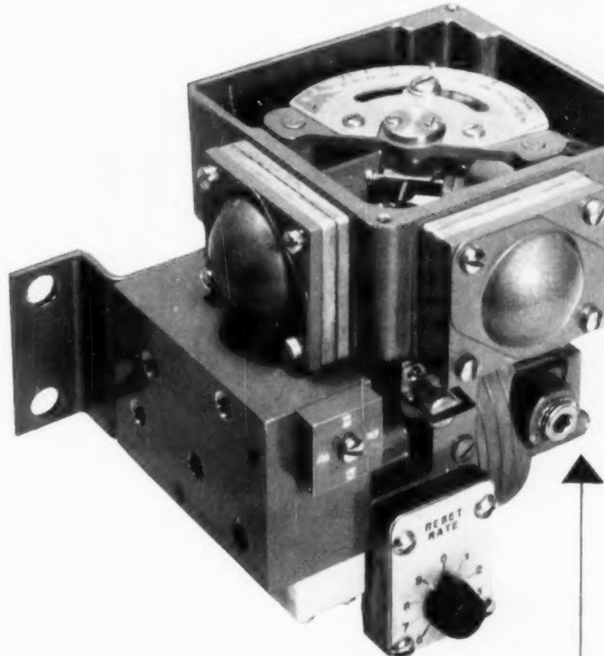




Base Any Of These...







ON THIS

For accurate, efficient, dependable control systems— new Rockwell-Built REPUBLIC TYPE "VC" PNEUMATIC CONTROLLER

Here is an all purpose null-balance-vector controller for use with any pneumatic transmitter. Its proportional band ranges from 2% to 500% *without changing parts*, for quick adaptation to changes in process requirements. Especially important in high-precision control, Republic's Type VC has exceptional sensitivity and a narrow dead band (less than 0.05%). Its high capacity non-bleed pneumatic amplifier consumes little air, keeps output ample. A selector block permits reverse or direct action. Local or remote pneumatic set point optional.

Companion instruments—using an identical null-balance-vector "heart"—include differential pressure, temperature and pressure transmitters . . . ratio, totalizing, multiplying, squaring and square - root - extracting relays. Many parts are interchangeable among the instruments in this "family". Besides reducing spare parts inventory, the similarity of components simplifies the task of training personnel.

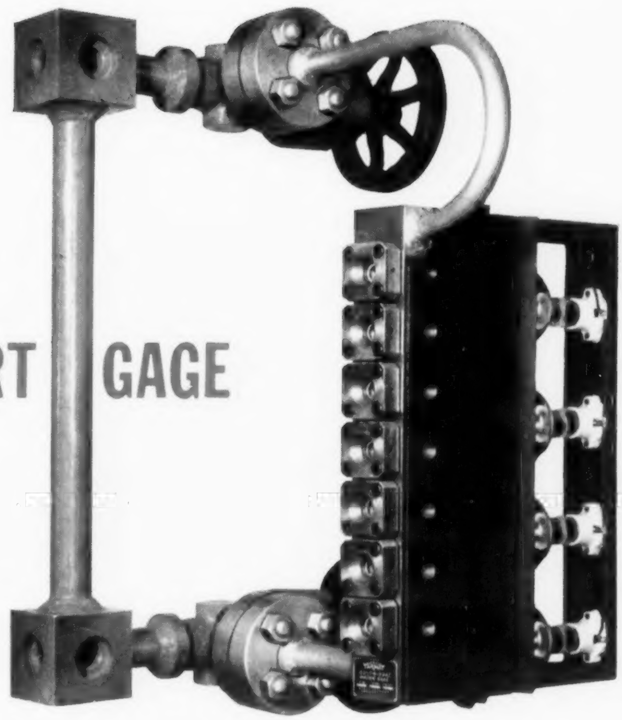
Let a Republic engineer show you how these instruments can help to achieve accurate, efficient, dependable control systems. Sales offices in principal cities throughout the U.S. and Canada. Write to Republic Flow Meters Company (subsidiary of Rockwell Manufacturing Company), 2240 Diversey Parkway, Chicago 47, Illinois. In Canada: Republic Flow Meters Canada, Ltd., Toronto.

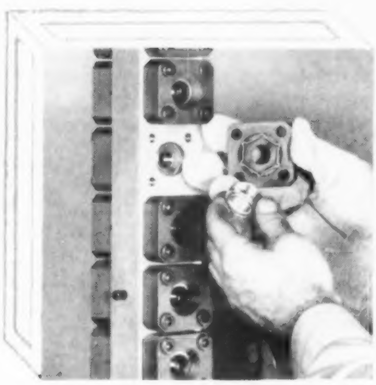
REPUBLIC INSTRUMENTS AND CONTROLS

more fine products by

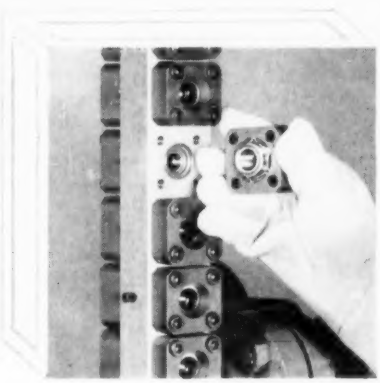
ROCKWELL

servicing the new YARWAY COLOR-PORT GAGE is easy as a b c





a Remove four cap screws and lift off cover assembly (held in right hand). Install new port assembly (glass-mica-gasket, shown in left hand) in cover. This port assembly is part of the complete cover assembly.



b Re-install complete cover assembly. Sealing gasket automatically seats in gasket groove in body.

c Tighten down four Allen cap screws with standard wrench (no torque wrench required).

New Yarway Color-Port Boiler Water Level Gages (for pressures to 3000 psi.) offer not only this new ease of maintenance but insure brilliant red and green readings of steam and water.

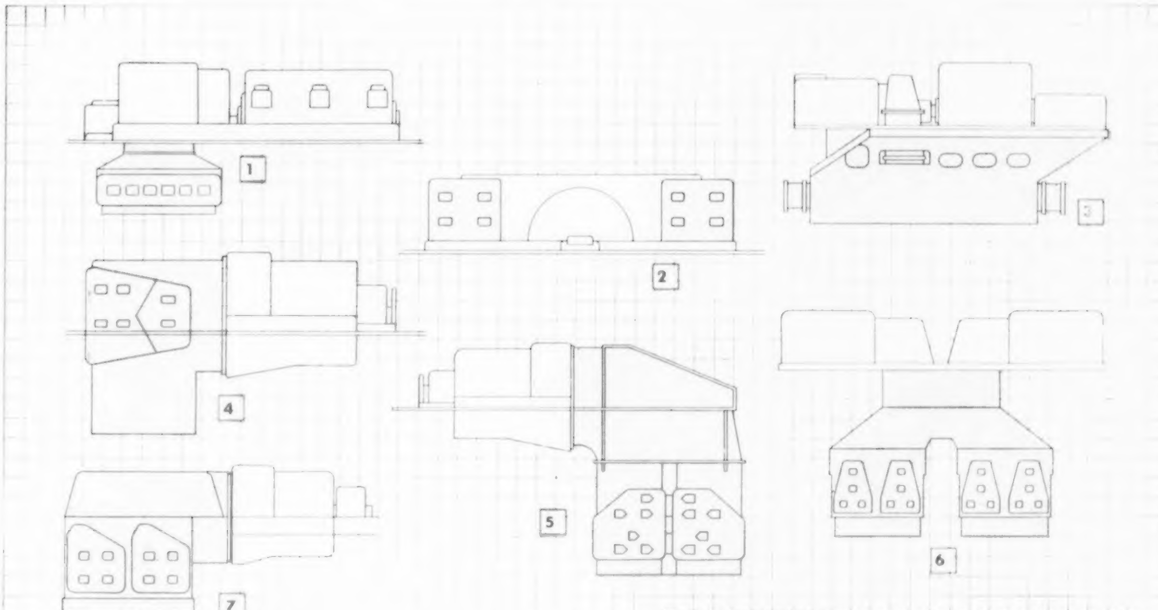
For full details, write for Yarway Bulletin WG-1814.

YARNALL-WARING COMPANY, 100 Mermaid Ave., Philadelphia 18, Pa.
Branch Offices in Principal Cities

GO YARWAY WITH CONFIDENCE

Look what Ingersoll-Rand is doing to surface condenser design

to help power companies produce more kilowatts in less space and with better efficiency



EACH of the surface condenser designs shown above didn't "just happen." To the contrary, each one was deliberately and painstakingly engineered by Ingersoll-Rand after careful study and analysis of the customer's requirements. And because the basic rectangular design of I-R condensers permits varying the ratios of length, width and height over wide limits, a great variety of space requirements can be effectively met . . . turbine-condenser arrangements can be "tailored" for each installation. Furthermore, this is accomplished without departing from the fundamental proven *high-performance* principles such as controlled longitudinal distribution of steam with controlled flow areas and velocities . . . principles which Ingersoll-Rand originated and developed to increase condenser efficiencies.

These advantages help to make I-R condensers the most efficient available—readily adaptable to all steam plant requirements. And you'll find that I-R condensers are faster and easier to erect, too, because most of the welding is done at the Ingersoll-Rand factory under close quality control.

Let an Ingersoll-Rand Engineer tell you more about the advantages and the engineering that has made I-R condensers first choice with so many power companies. Ask him for your copy of Form 9300, New Horizons in Condenser Progress.

1 **Low Headroom** — This I-R condenser provides 65,000 sq ft of surface in an under turbine area that allowed only 14′ 8″ of headroom.

2 **No Headroom** — This twin shell 110,000 sq ft condenser is integrated into the advanced design dual side-exhaust turbine.

3 **Integral Foundation** — The I-R condenser here serves as the foundation for the bottom-exhaust turbine and generator.

4 **Axial Exhaust Turbine** is served by this I-R condenser located at the turbine floor and direct connected for straight through steam flow.

5 **Low Syphon Head** was one of the requirements for this I-R condenser connected to the axial-exhaust turbine by a precisely engineered transition elbow.

6 **Independent operation** of either shell and reduced headroom result from this twin shell arrangement of a 212,000 sq ft I-R condenser.

7 **Compact, space saving** arrangement of a 110,000 sq ft condenser connected to an axial-exhaust turbine with steam flow to the condenser tube bundles located offset and below the turbine exhaust centerline.

Ingersoll-Rand

11 Broadway, New York 4, N. Y.

COMPRESSORS • TURBO-BLOWERS • ROCK DRILLS
AIR TOOLS • CENTRIFUGAL PUMPS
CONDENSERS • GAS AND DIESEL ENGINES

There's no substitute for experience in engineered products

POWELL

world's largest family of valves

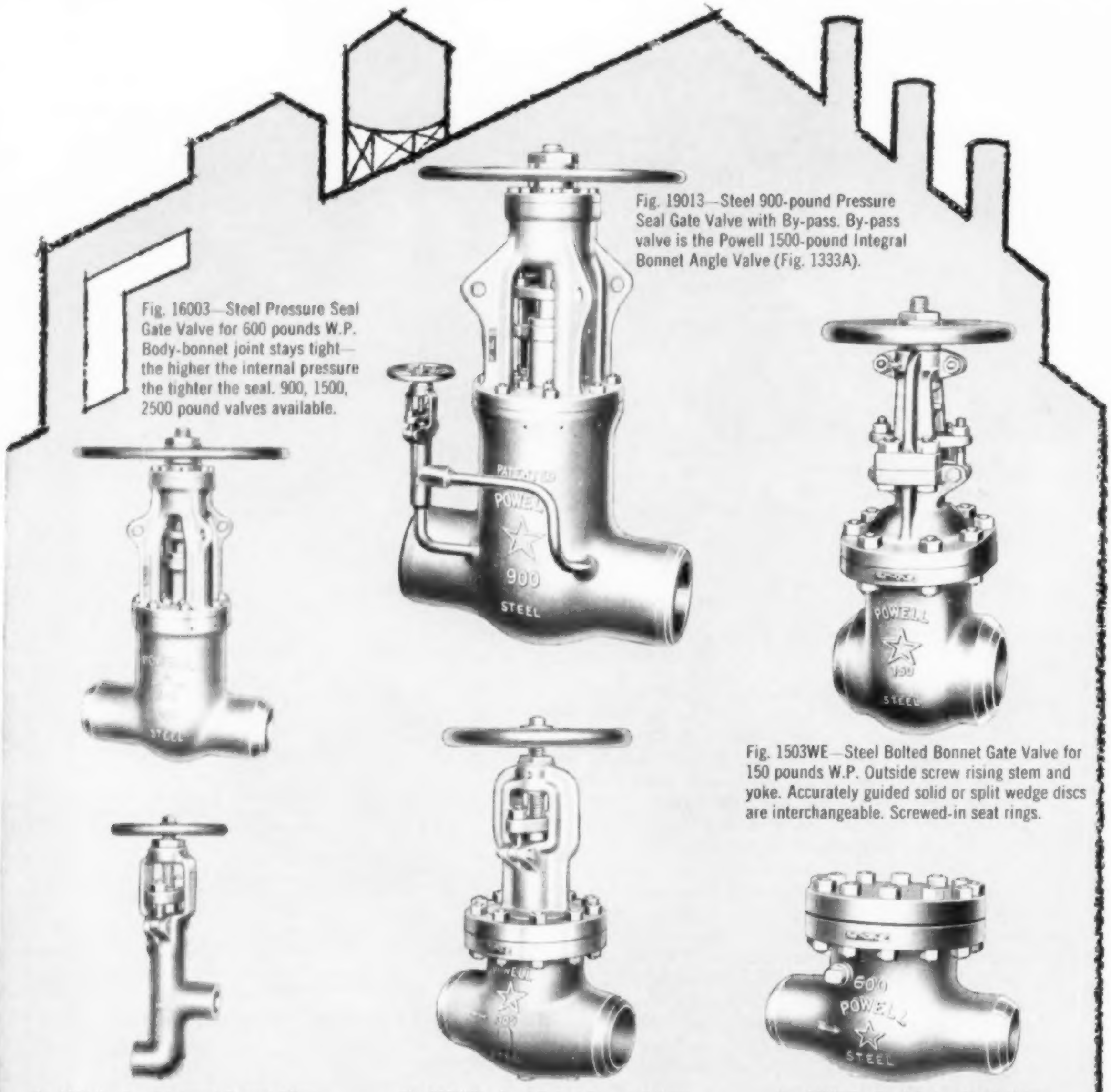


Fig. 19013—Steel 900-pound Pressure Seal Gate Valve with By-pass. By-pass valve is the Powell 1500-pound Integral Bonnet Angle Valve (Fig. 1333A).

Fig. 16003—Steel Pressure Seal Gate Valve for 600 pounds W.P. Body-bonnet joint stays tight—the higher the internal pressure the tighter the seal. 900, 1500, 2500 pound valves available.

Fig. 1503WE—Steel Bolted Bonnet Gate Valve for 150 pounds W.P. Outside screw rising stem and yoke. Accurately guided solid or split wedge discs are interchangeable. Screwed-in seat rings.

Fig. 1331-A—Small Integral Bonnet Offset Globe Valve for 1500 pounds W.P. One-piece construction, light, compact. Stellite hard faced seat and disc assure long service.

Fig. 3031WE—Steel Bolted Bonnet O.S.&Y. Globe Valve for 300 pounds W.P. Can be supplied with plug type discs for either steam or oil service. Screwed-in seat rings.

Fig. 6061WE—Steel Swing Check Valve for 600 pounds W.P. Heavily bolted cap. Provides straight full flow through the valve when disc is in open position.

For every flow control problem Powell offers more kinds or types, available in the largest variety of metals and alloys to handle every flow control requirement. Powell distributors are located in all principal cities and maintain inventories to fill almost any need. For special engineering problems, write direct to:

THE WM. POWELL COMPANY • *Dependable Valves Since 1846* • **Cincinnati 22, Ohio**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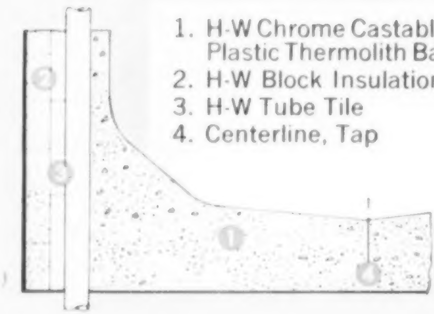
SPECIAL
ABOUT
LJUNGSTROM'S®
easy maintenance features
Ljungstrom maintenance is fast and easy because it, too, is a "designed-in" function of the preheater. Necessary work has been foreseen by such money-saving features as:
• Self-cleaning action that loosens deposits by expansion and contraction—coupled with high-velocity soot blowers for daily in-operation purging.
• Cold-end heating elements are factory packed—can be easily replaced, or reversed when one edge starts to thin...which is vital for cold end service and essentially doubles surface life.
• Large inspection port and vapor-proof light to permit observation of heating surface, even during operation.
• Trained service engineers make periodic visits and are available on short notice for special problems.
That's why seven out of ten installations are Ljungstrom. For the full story write for our 38-page manual.
The Air Preheater Corporation, 60 EAST 42ND STREET, NEW YORK 17, N. Y.

In recovery boiler furnaces . . .

HARBISON-WALKER CHROME REFRACTORIES

continue to make outstanding service records

ALL-MONOLITHIC CONSTRUCTION



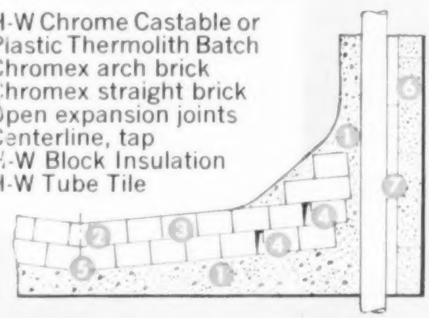
1. H-W Chrome Castable or Plastic Thermolith Batch
2. H-W Block Insulation
3. H-W Tube Tile
4. Centerline, Tap

H-W CHROME CASTABLE (dry in sacks)

At many mills it is considered the best refractory yet developed for bottoms and lower side walls of paper mill recovery furnaces. It has high density and is extremely resistant to chemical attack and erosion by the molten smelt. It is easy to use without special experience, and may be applied by air-placement guns and by casting both for complete installations and for maintenance patching.

BRICK AND MONOLITHIC CONSTRUCTION

1. H-W Chrome Castable or Plastic Thermolith Batch
2. Chromex arch brick
3. Chromex straight brick
4. Open expansion joints
5. Centerline, tap
6. H-W Block Insulation
7. H-W Tube Tile



PLASTIC THERMOLITH BATCH (Plastic Chrome Ore)

This leading plastic chrome ore refractory is a ramming mixture which becomes hard, dense and highly impervious to the molten salts, upon air drying. For the monolithic lined parts of recovery furnaces, it provides the high, all-temperature strength and resistance to chemical corrosion, required for long dependable service.

CHROMEX BRICK

High mechanical strength, constancy of volume at high temperatures, resistance to spalling and to chemical attack by sodium salts are outstanding properties which account for the excellent service records of Harbison-Walker CHROMEX. This chemically-bonded chrome-magnesite refractory is especially suited for the linings of all designs of recovery units.

THERMOLITH Bonding Mortar

(in sacks or steel drums)

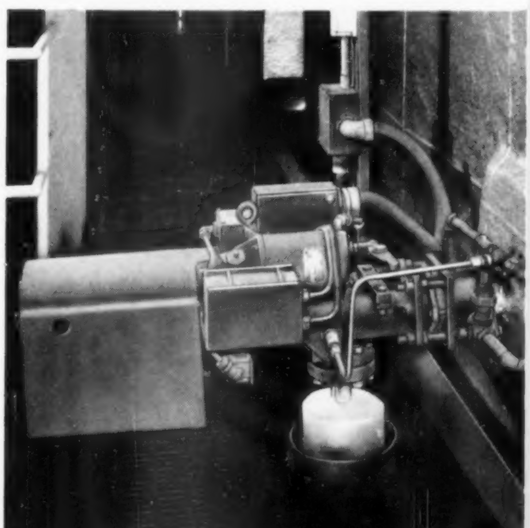
Made with a chrome ore base, this cold setting cement provides a strong, highly impervious joint compatible in thermal expansion and chemical composition with CHROMEX brick.

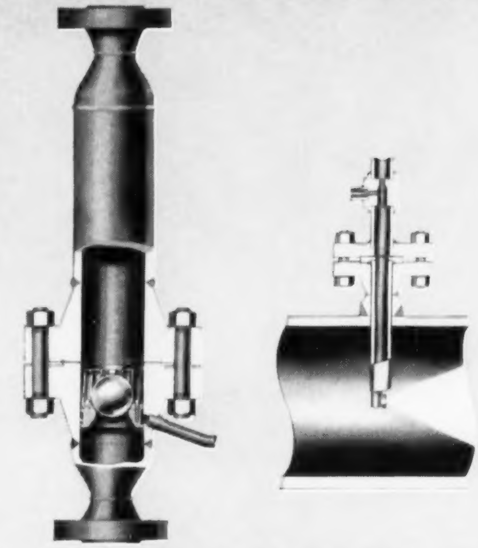
HARBISON - WALKER REFRACTORIES COMPANY
AND SUBSIDIARIES
World's most complete refractory service — with all types and classes
General Offices Pittsburgh 22, Pennsylvania

C-V NEWS NOTES

Finger tip control for Gannon's operators . . . with Vulcan's Automatic-Sequential group and unit control. Operator can preselect any number of soot blowers for automatic sequence . . . preselect any number of units or groups of units to be operated individually . . . or switch instantly to single unit operation. Program lights and toggle switches are located on engraved boiler diagram. Automatic-Sequential systems use steam, air or a combination as the blowing medium without a change in equipment. Write for Bulletin 1029.

Wall deslagger conserves steam . . . minimizes average slag thickness. Vulcan's RW-3E is equipped with dual motors: one to traverse *fast* to get the nozzle to and from the blowing position almost instantly . . . the other to rotate the tube slowly for thorough cleaning. The RW-3E may be installed indoors or out, at almost any angle from side wall, roof or floor. It has only one outside stuffing box, no threads or working parts in the blowing medium to assure easy maintenance. Write for Bulletin 1034.

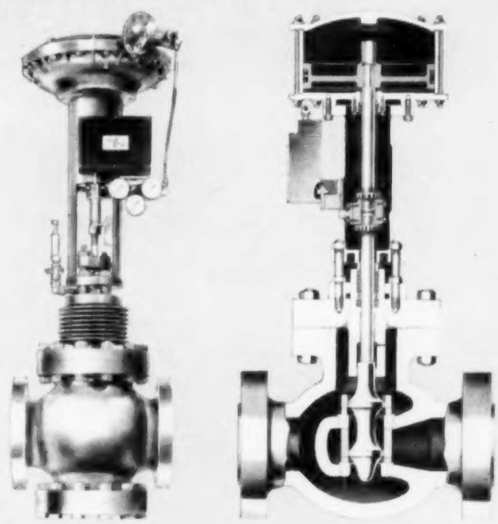




Desuperheaters improve temperature control. Variable-Orifice Desuperheater* (left) uses a weighted steel ball to control orifice opening . . . holds reduced steam temperature constant only twenty feet downstream from desuperheater outlet, even over a 50-to-1 load range. Bulletin 1037.

*Patent Applied For

Steam-Assist (right) meets all specifications for conventional steam-atomizing type yet normally uses assisting steam only on light loads where control is most difficult. Mechanical-atomizing types also available. Bulletin 1024-A.



Versatile regulator valves offer new simplicity of design. Diaphragm-type CV-D (left) is designed for remote control service. It can be direct or reverse acting, has excellent rangeability. Piston-type CV-P (right) is designed for high-duty service . . . assures maximum power with precise positioning.

All Copes-Vulcan valves are tailored to the job. The style of valve port is selected to provide optimum control for specified operating conditions. Bulletin 1027.

From pit to plant, LeTourneau—

New 400 HP LW-80—world's largest coal hauler—is 12' high by 12' wide, 54½' long. It hauls 100 cubic yards (80 tons) per trip at speeds up to 34 mph.

Westinghouse moves with coal

Earthmoving equipment manufacturer estimates steam generated from coal costs only half as much as competitive fuels

At LeTourneau-Westinghouse Co., Peoria, Ill., steam is used to heat an area that has swelled to a million square feet of space. Increasing demand created the need for a modern heating plant to replace obsolete equipment. When plans were drawn for the new plant, fuel surveys indicated that the firm should continue to use coal for availability and economy. Estimates showed that steam produced from burning coal in this area costs about *half as much* as from competitive fuels.

Today LeTourneau-Westinghouse has found that burning coal in modern equipment cuts costs in several ways. Fuel costs, of course, are at a minimum. Automatic, efficient operation has lowered coal and ash handling costs. And manpower requirements have been reduced from 12 to 7 men.

Facts you should know about coal

You'll find that bituminous coal is not only the lowest-cost fuel in most industrial areas, as in the case of LeTourneau-Westinghouse, but up-to-date coal burning equipment can give you 15% to 50% more steam per dollar. Today's automatic equipment can pare labor costs and eliminate smoke problems. And vast coal reserves plus mechanized production methods mean a constantly plentiful supply of coal at stable prices.

Technical advisory service

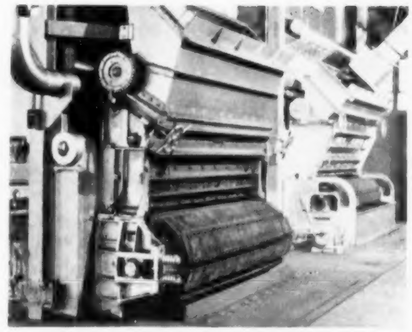
To help you with industrial fuel problems, the Bituminous Coal Institute offers a free technical advisory service. We welcome the opportunity to work with you, your consulting engineers and architects. If you are concerned with steam costs, write to the address below. Or send for our case history booklet, complete with data sheets. You'll find it informative.

Consult an engineering firm

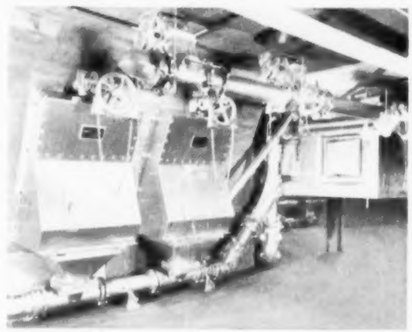
If you are remodeling or building new heating or power facilities, it will pay you to consult a qualified engineering firm. Such concerns—familiar with the latest in fuel costs and equipment—can effect great savings for you in efficiency and fuel economy over the years.

BITUMINOUS COAL INSTITUTE

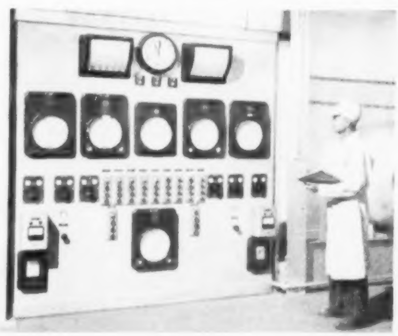
Department C-09
Southern Building, Washington 5, D. C.



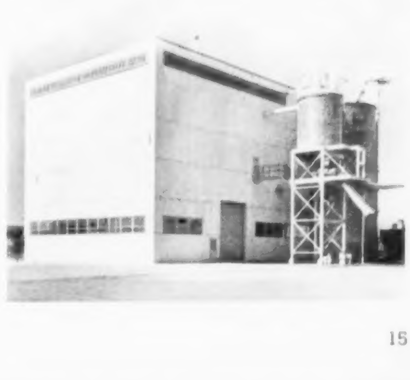
View in the LeTourneau-Westinghouse heating plant. In foreground is an Erie City 40,000 lb/hr type I.W.T. boiler which is used on stand-by basis and for peak loads in extremely cold weather. At the right rear is Erie City 60,000 lb/hr main boiler. Both are fired by Combustion Engineering Inc. traveling grate stokers.



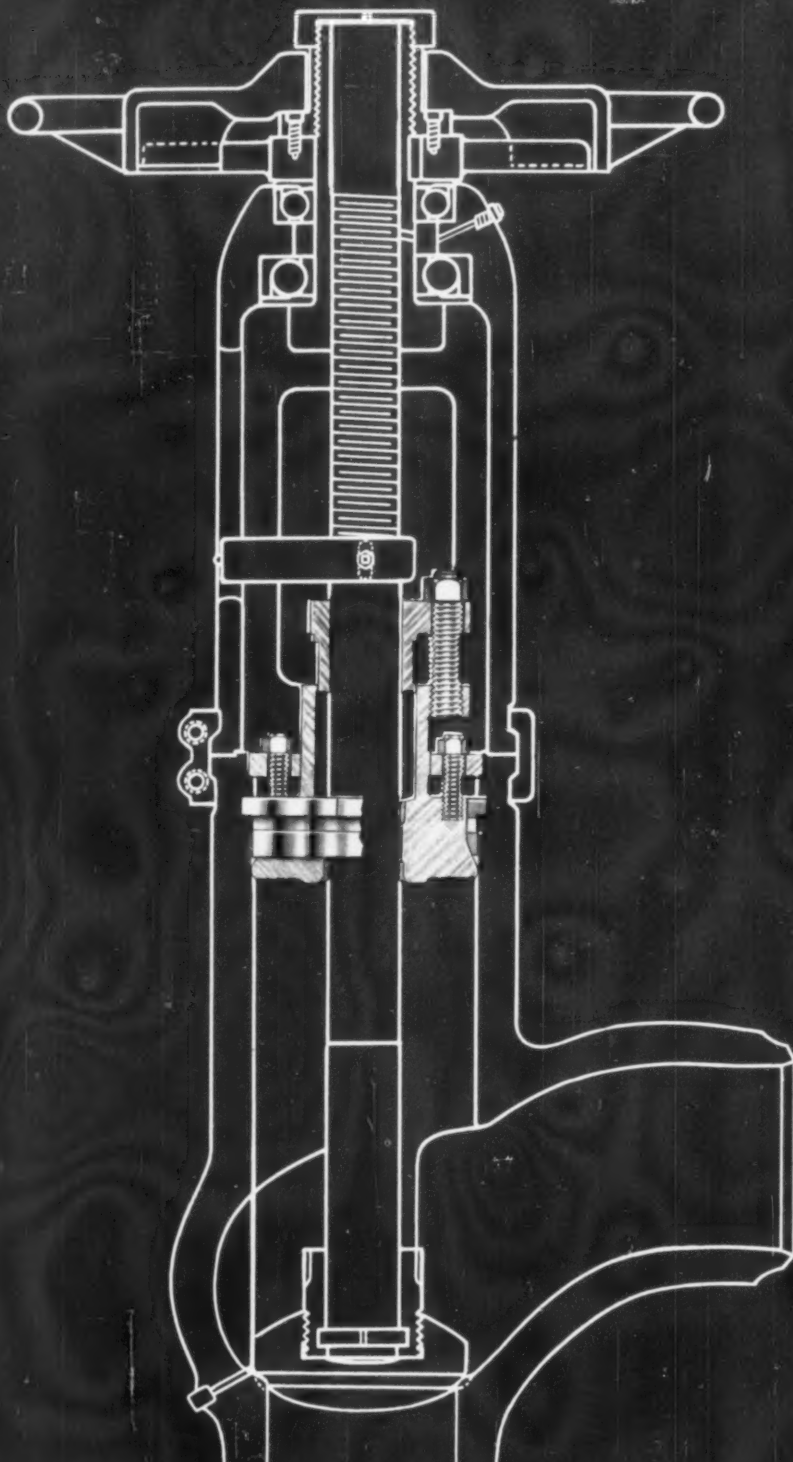
Shown are ash hoppers, part of United Conveyor ash handling system. Ashes are dumped from ash pits into hoppers from where they are piped pneumatically to silo. This system also handles dust from Prat-Daniel dust collectors.



Bailey Control Panel regulates both boilers, including boiler meters and recorders, feedwater controllers and alarms, drum level recorders and alarms, multipointer gages for draft control and return condensate recorders.



New heating plant showing 9000 cu ft Kalamazoo Tank & Silo Co. coal silo (on right). Coal feeds from railroad car into track hopper, then moves by screw conveyor to 40 ton/hr bucket conveyor which elevates coal to top of silo. Entire coal handling system by Beaumont-Birch Co. To left of coal silo is Kalamazoo 25-ton ash silo equipped with United Conveyor dustless rotary unloader.



EDWARD PRESSURE-SEAL ANGLE VALVE

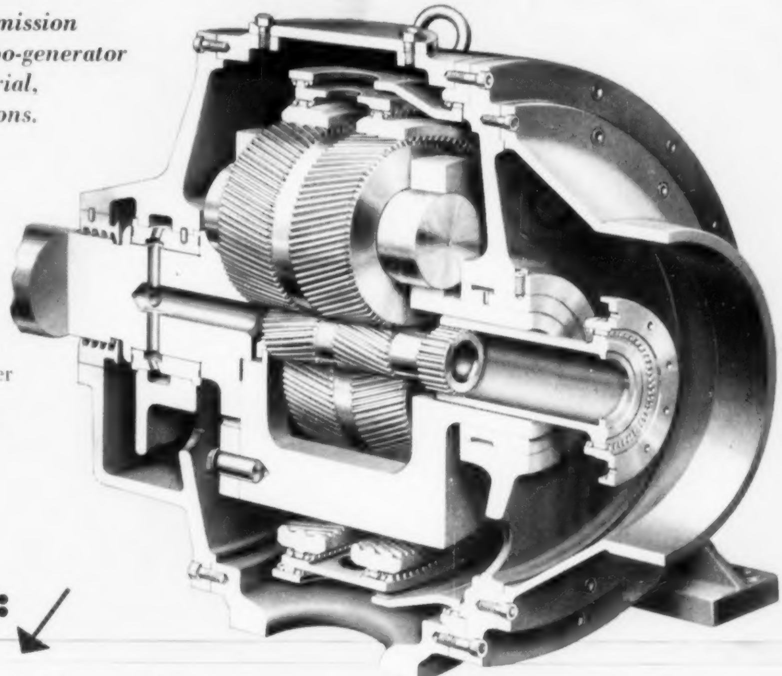
DE LAVAL-STOECKICHT
PLANETARY GEAR

...for high speeds...high horsepower

Proved in hundreds of installations abroad totalling over 3,000,000 horsepower—now available in America!

For all high torque power transmission applications such as pump turbo-generator and compressor drives in industrial, municipal and marine installations.

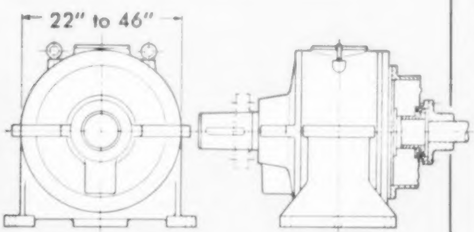
This cutaway view of the De Laval-Stoeckicht Planetary Gear shows how it provides flexibility for proper load distribution throughout the gear members. The thoroughly proved and tested design is completely reliable in transmitting high horsepower for high speed applications. • Highest efficiencies (98% or higher) ... no high speed bearings ... less friction losses.



Check These Advantages:

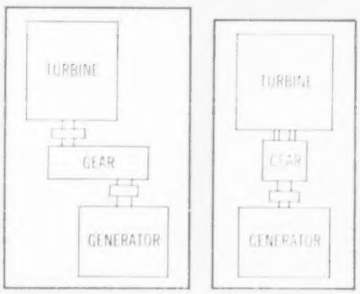
Small Size—Light Weight

Compact—low weight per hp. Sizes range from 22" to 46" in diameter, depending on horsepower requirements. Example: 5000 hp planetary unit weighs 1700 lbs. against 6000 lbs. for conventional gear.



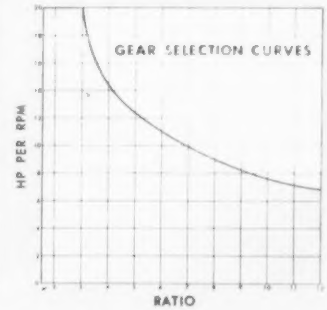
Convenient Arrangement

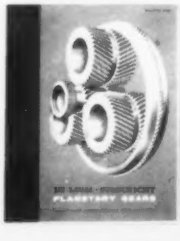
Co-axial or "in-line" arrangement of gear members takes up far less space than parallel axis gears of equivalent horsepower rating.



Wide Application

Capacity range shown in shaded area on chart below. For other applications, contact your De Laval Sales Engineer.





For further details, write for Bulletin 2400.

DE LAVAL *Steam Turbine Company*

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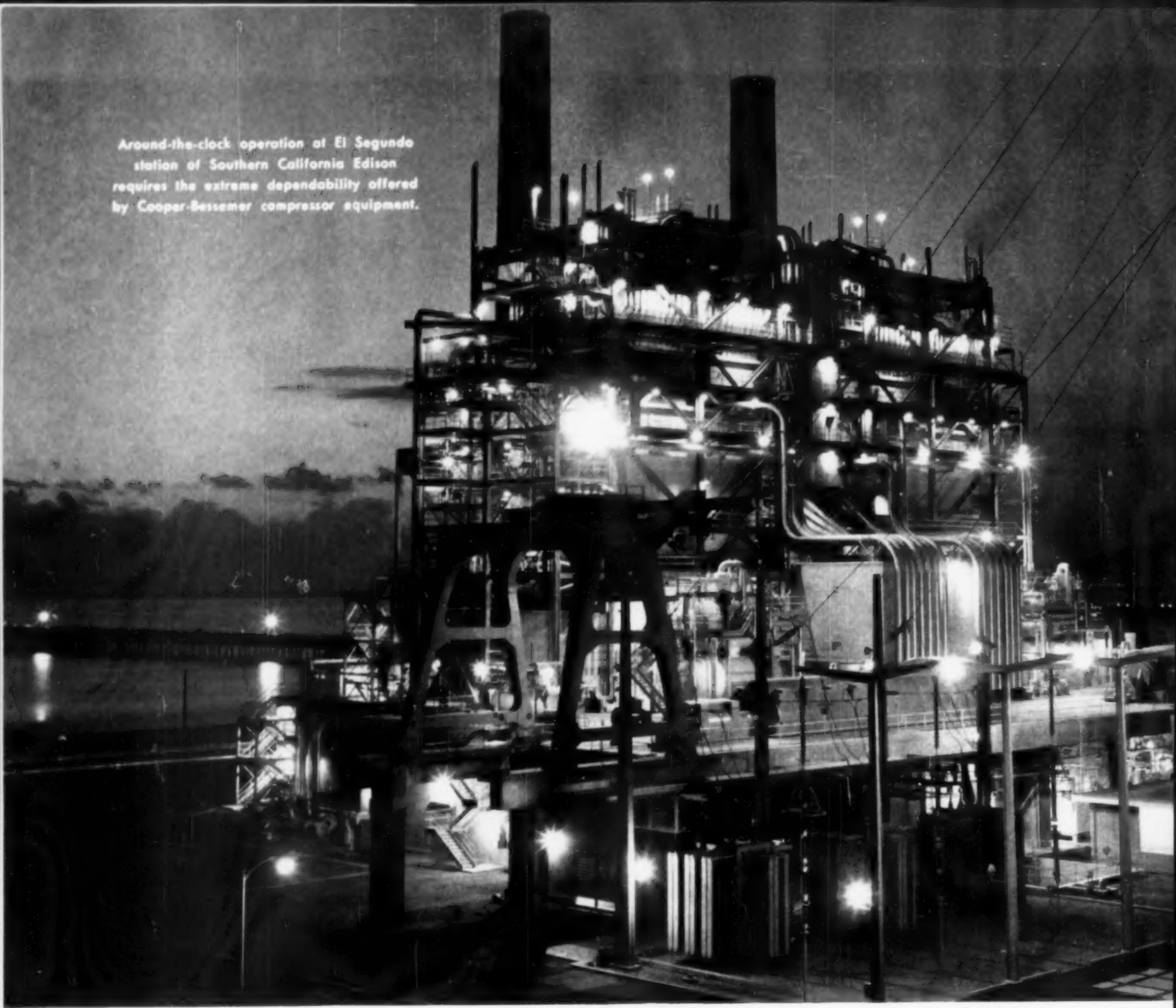

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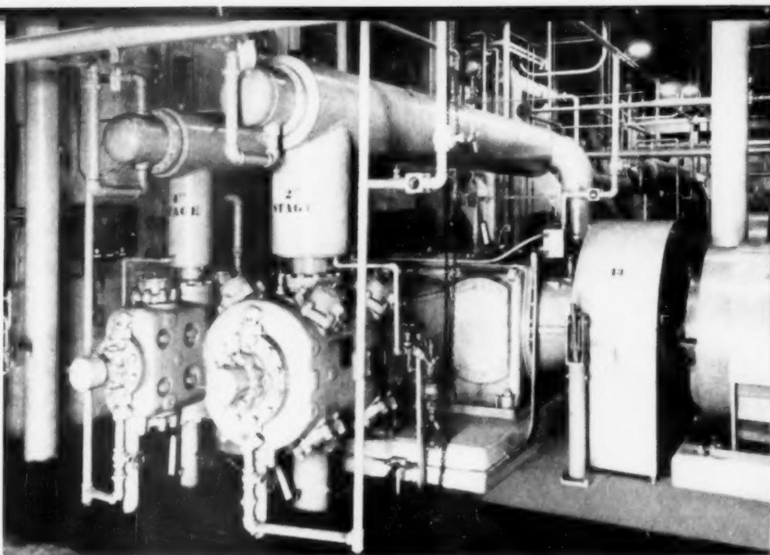
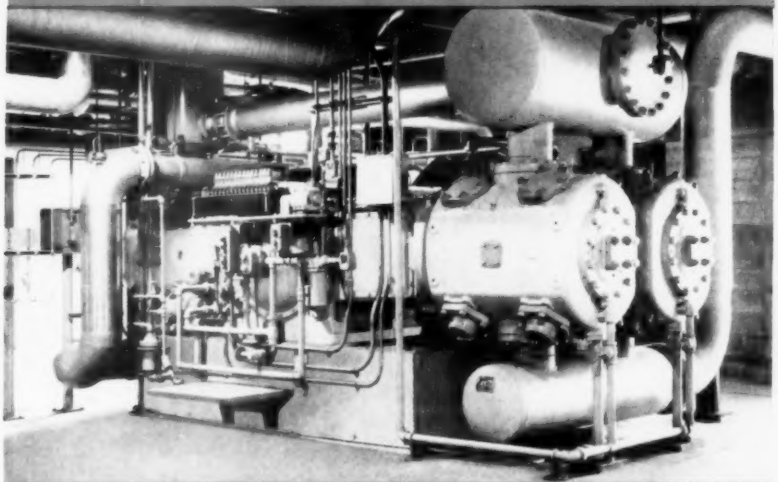


Extreme accessibility is another plus feature of motor-driven FM compressors in the Sayreville station of Jersey Central Power & Light.



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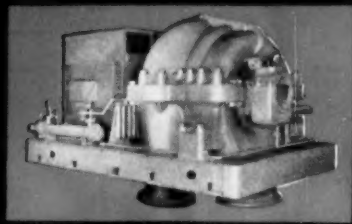

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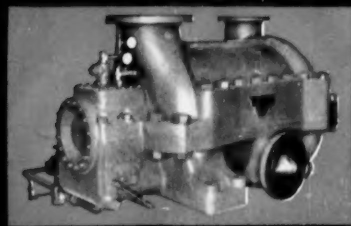

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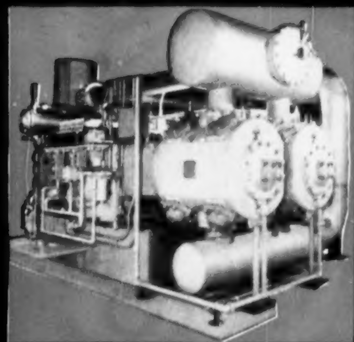

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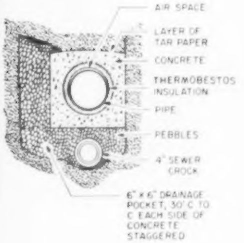
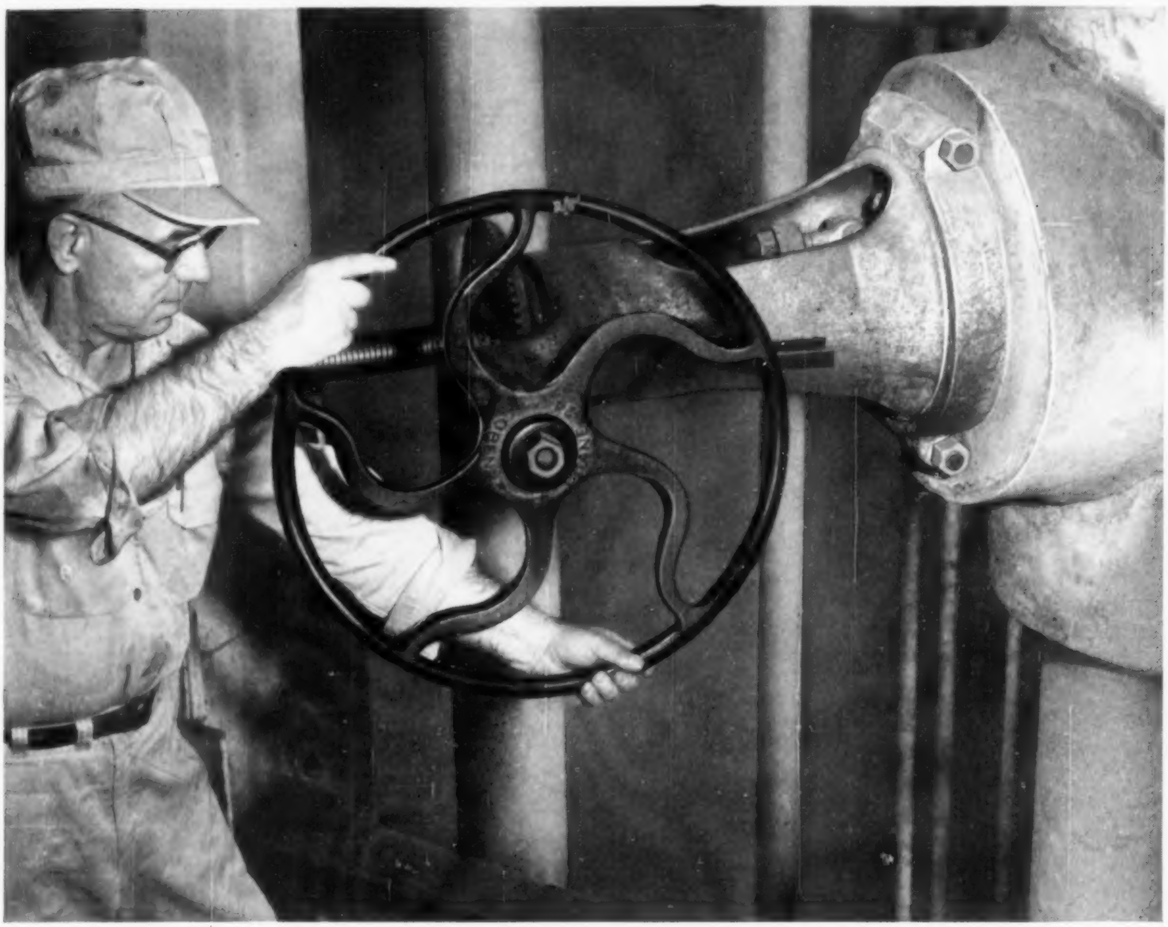


Above, Solid-Pour Steam Main Conduit

Ready for "solid pour." Thermobestos insulation is shown in position with waterproof jacketing partly installed (see diagram for trenching detail).

Easily withstands weight of poured concrete. Here in the same trench Thermobestos is shown with concrete poured solidly around it.

Refinery Power Plant Compares Bonnet Joints



No maintenance for Crane Pressure-Seal valve on 10-year high-pressure/high-temperature service

Ten years ago two 10-inch, high-pressure-temperature gate valves were installed in the power plant of a Midwest refiner. Each one was installed on a boiler lead line, where steam service is 900° F. at 1280 psi.

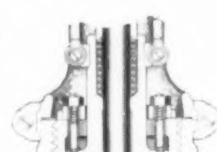
One valve is a conventional bolted bonnet type; the other, a Crane Pressure-Seal Bonnet Valve.

The bolted bonnet valve has required major maintenance four times in ten years because of bonnet joint leakage.

The Crane Pressure-Seal Bonnet Valve, during the same period, has never leaked . . . has needed no maintenance of any kind!

Conventional valves fight line pressure with heavy flanges and bolts at the bonnet joint. The bolting requires restressing and maintenance with each extreme change in temperature. Crane Pressure-Seal design harnesses line pressure to keep the bonnet joint tight despite extreme variations in temperature and pressure.

For complete information about Crane Pressure-Seal Bonnet Joint Valves—gates, globes, angles, stop-checks and swing-checks—in 600, 900, and 1500-pound classes—call your Crane Representative, or write to address below for new Circular AD-2336.



Here's the Crane Pressure-Seal Joint that eliminates bonnet joint leakage and maintenance on high-pressure-temperature services.

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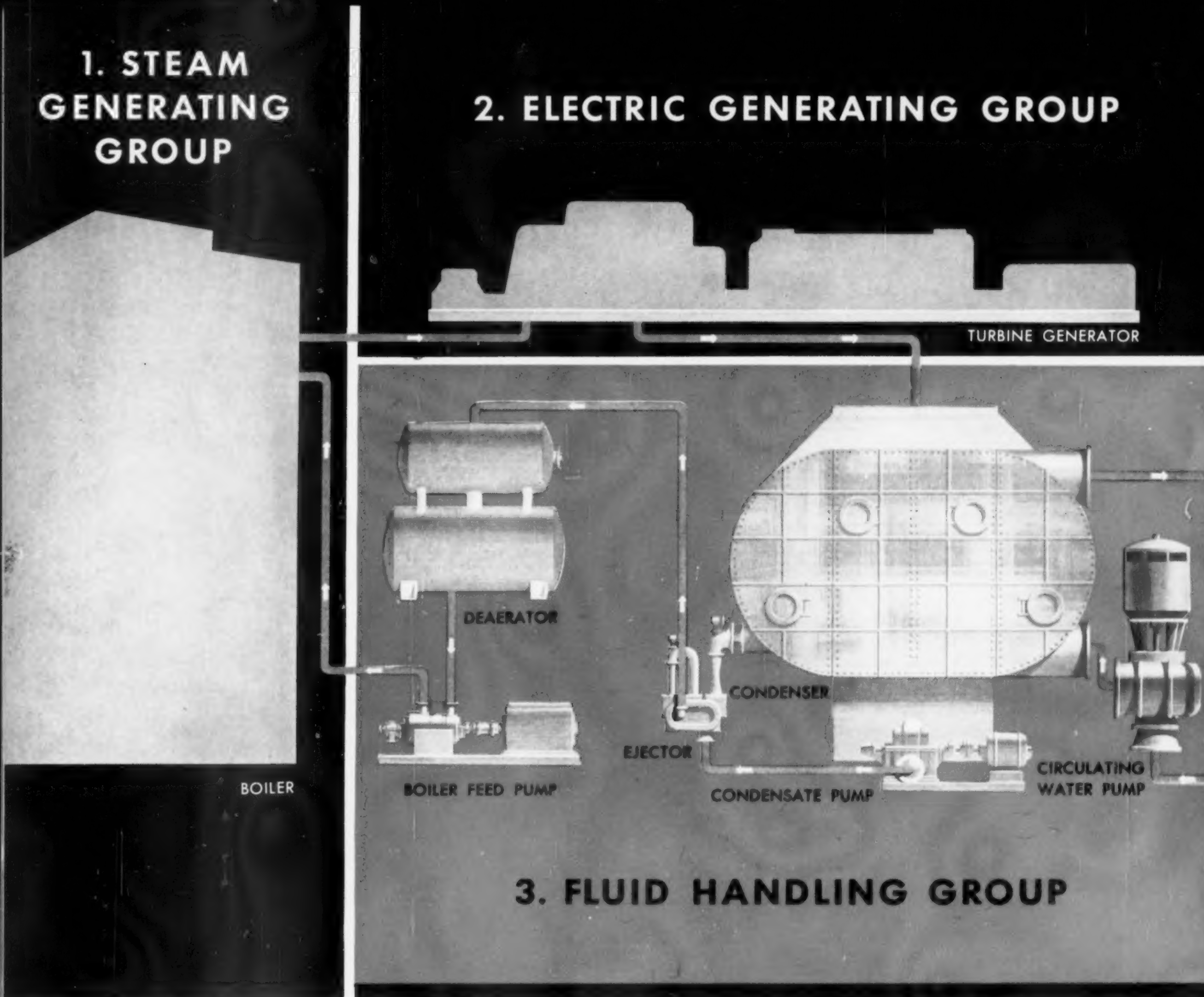

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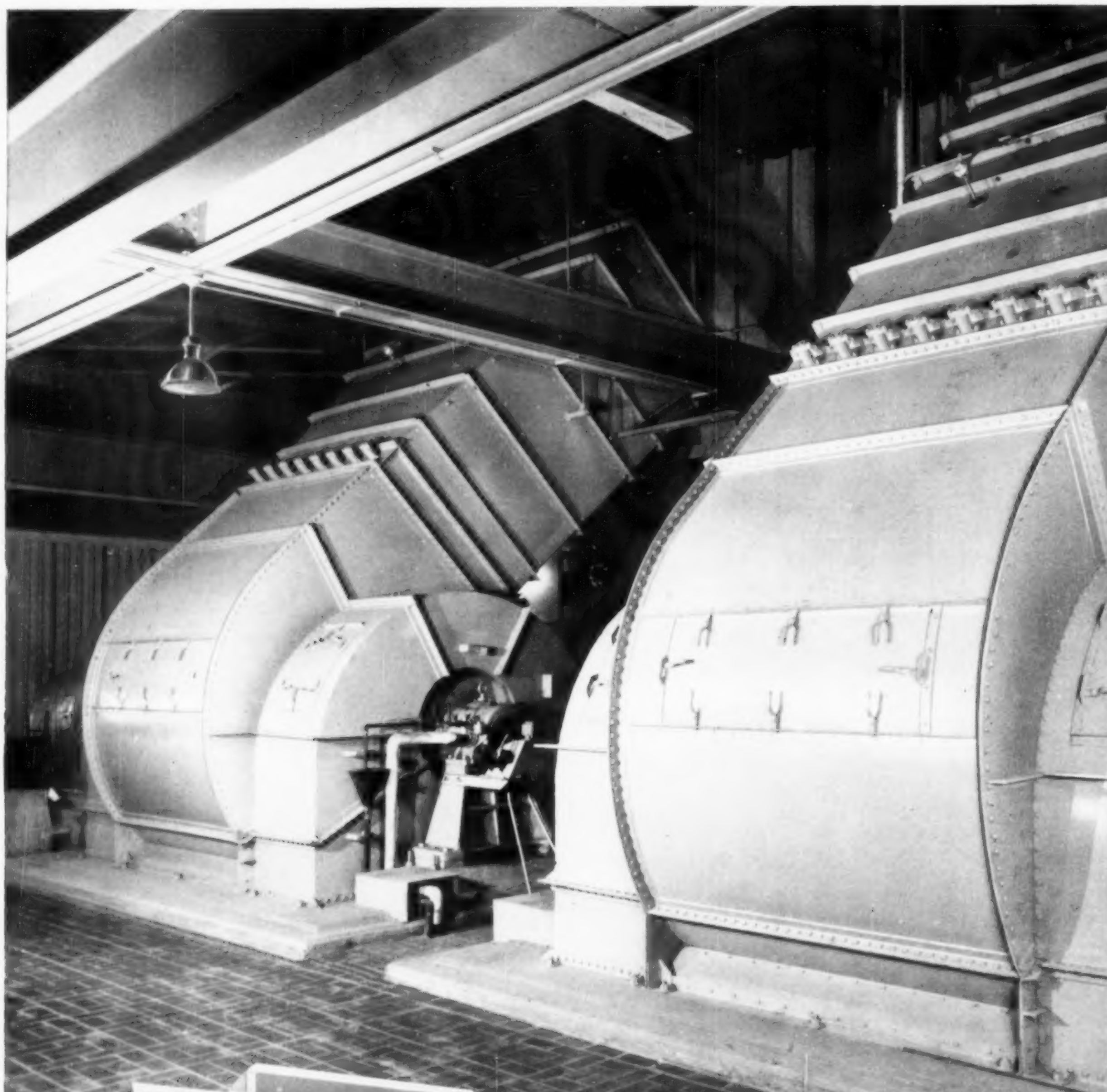

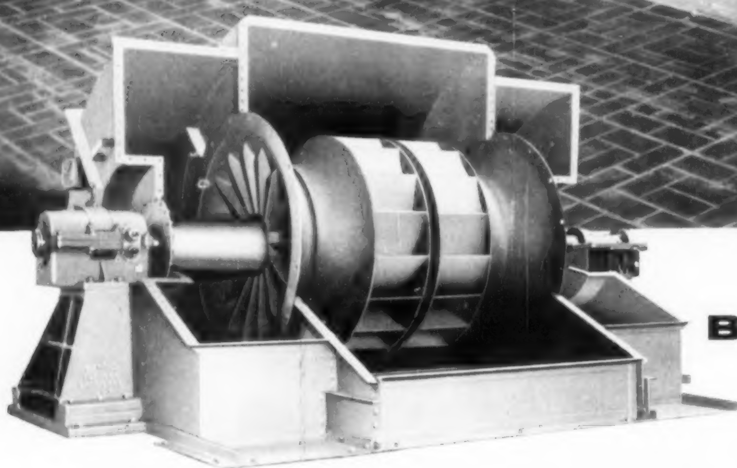

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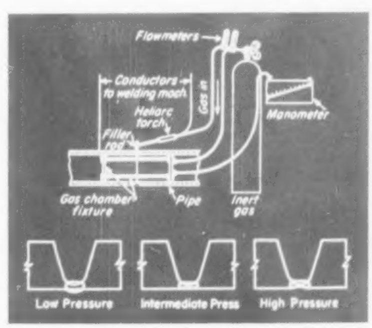

K-Weld root bead is deposited while pipe interior is under controlled inert gas pressure in specially designed portable gas chamber fixture. Diagram shows details of fixture, and how interior weld surface can be controlled by varying pressure.

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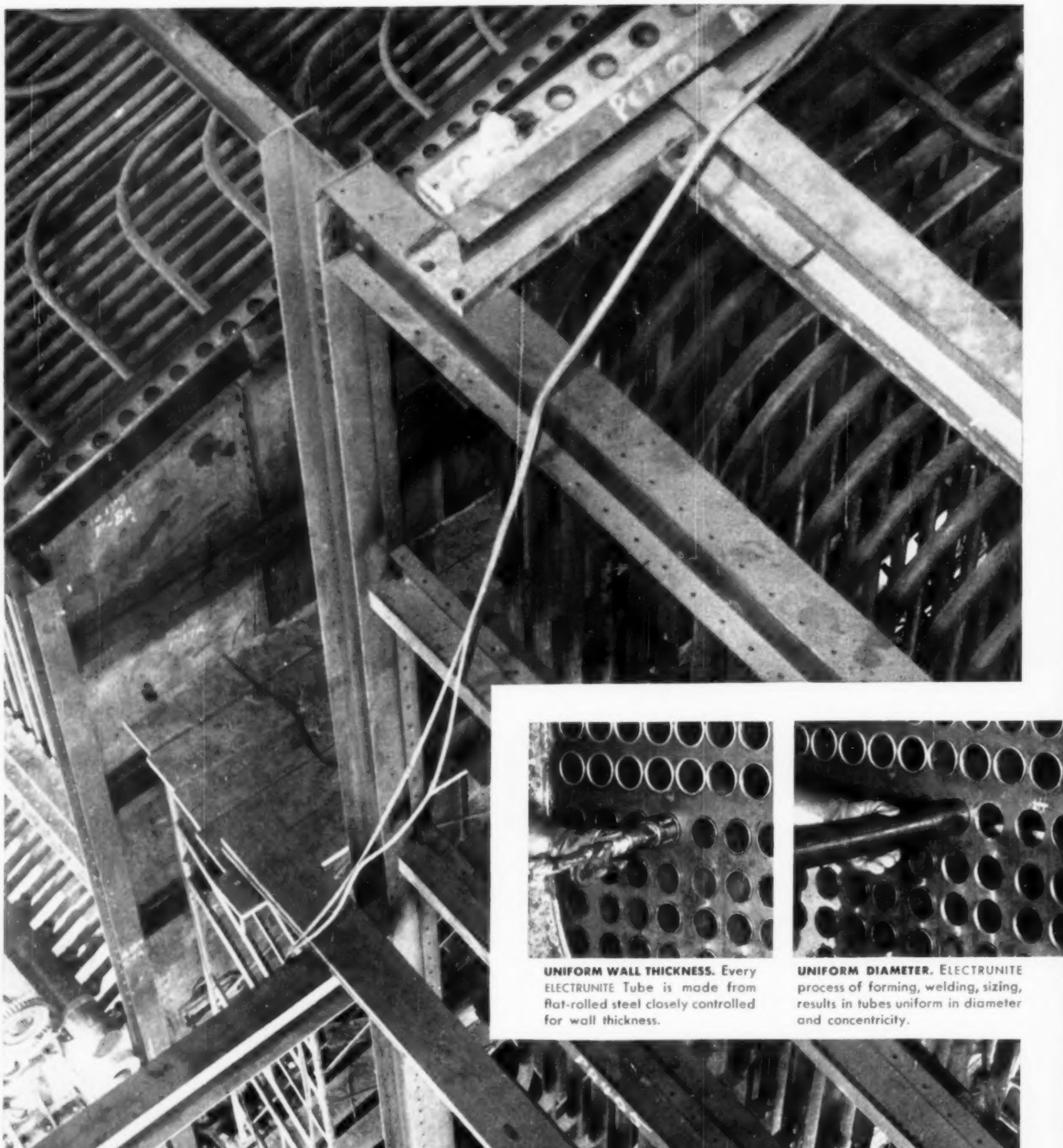

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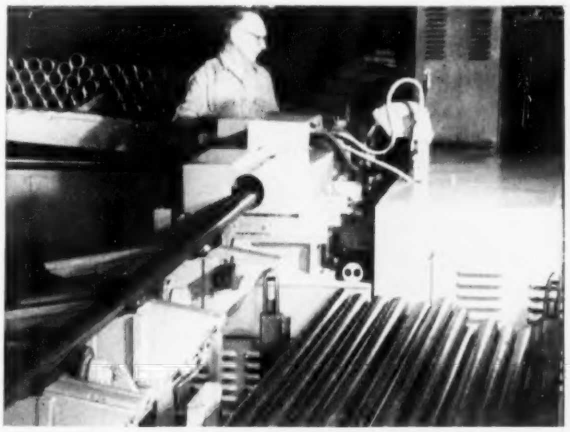

Wall Thickness	Minor Dimension of the Defect (Length or Depth)	Length X Depth Area in Square Inches
20 ga.	.006″	.0025″
18 ga.	.006″	.003″
16 ga.	12½% of Wall	.003″
14 and 13 ga.	12½% of Wall	.004″
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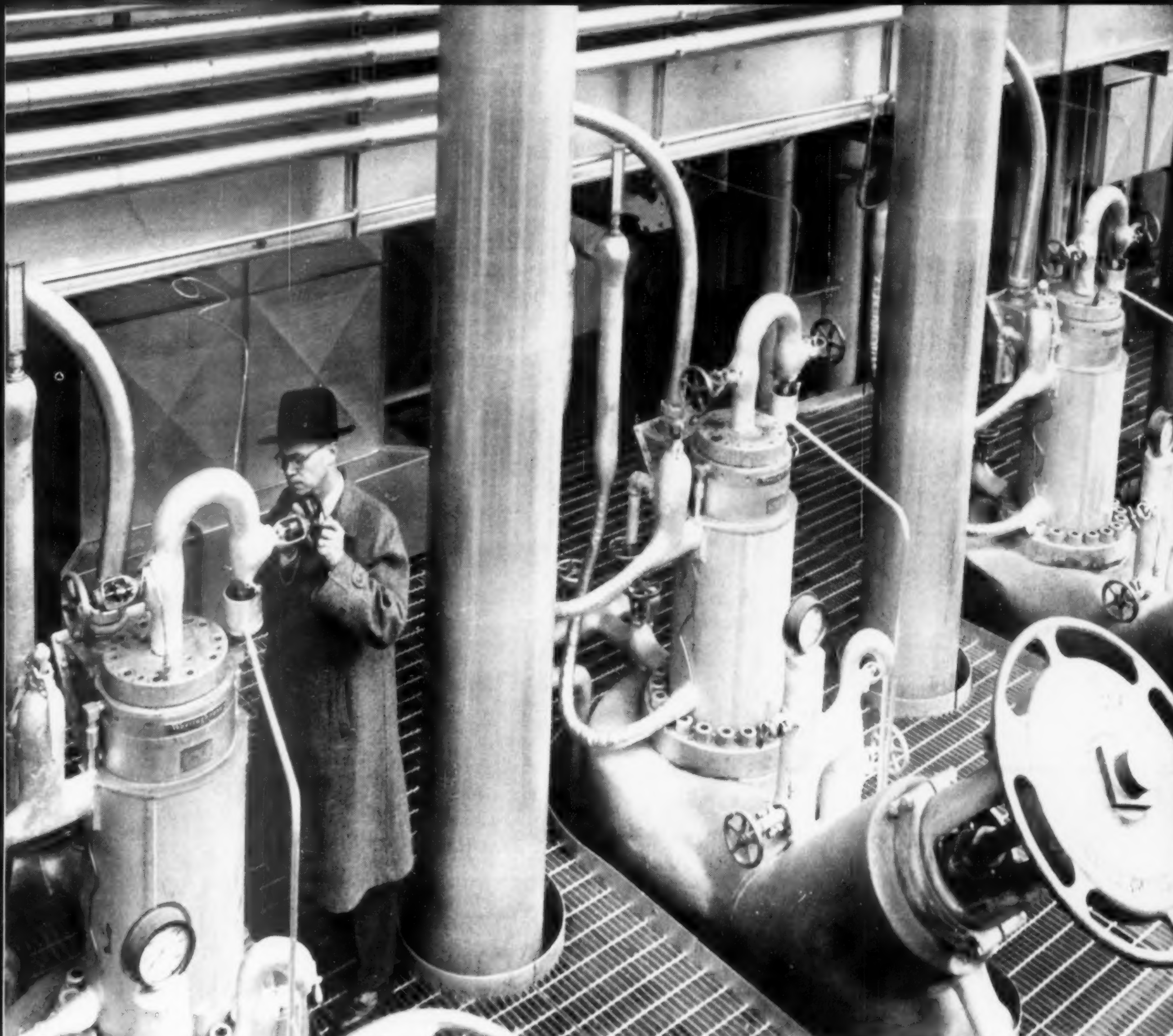

Westinghouse canned motor-pumps

specified for KP&L outdoor installation

The Kansas Power and Light Company recently increased the capacity of its Tecumseh Station with the addition of a 75,000-kw outdoor unit. This new addition, the most efficient in the plant, will operate day and night as the base load unit. To insure reliability, Westinghouse canned motor-pumps were specified for the Combustion Engineering Inc. controlled circulation boiler.

Westinghouse pumps were selected because: they have been proven for long-term continuous operation in many other installations; the zero-leakage feature eliminates injection water sealing. Two pumps provide the required boiler water circulation to deliver 635,000 pounds of 1000 F steam at 1800 psi. A third Westinghouse pump is kept on hot standby. All three are rated at 4070 gpm, 1958 psi, 615°F.

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COMBUSTION

Editorial

What About a History of Steam Power Engineering?

The July issue of *Mechanical Engineering* included a fascinating review of a recent book entitled "History of Hydraulics" by Hunter Rouse and Simon Ince. Among other things the reviewer, W. J. Rheingans, pointed out that the authors succeed in bringing alive, placing in proper perspective and giving meaning to the time-honored pioneers whose names are identified with the laws and formulae of hydraulics.

There are many books which present a superficially descriptive history of important developments in steam power engineering. Some of these are very handsomely illustrated, are written to attract youth to the excitement of engineering accomplishments, and serve a very useful purpose to awaken interest in the profession. Histories of the stationary steam engine and the steam locomotive are included in literally hundreds of volumes. Many manufacturers have prepared histories of their particular commercial achievements. Some engineering treatises, such as Stodola's two volume exposition on turbines, contain much valuable historical information.

Why then, you ask, propose adding another book to library shelves? The answer is that past histories have been more devoted to *things* than to men, to chronological setting forth of developments rather than to the impact of these developments upon society.

It is known to a few that the Perkins family of engineers designed and operated small 2000-psig boilers in nineteenth century England. But how does one explain the engineering background of persons who conceived of things so far beyond the vision of their contemporaries? What kind of a person was William J. M. Rankine who nearly a century ago wrote, as a part of a book on steam engines, the first exposition of thermodynamics, a treatment that has been copied and juggled about by literally hundreds of writers of later thermodynamics texts? And what about the men around Thomas A. Edison in the early days of the central station industry? How did they combine technical pioneering with manufacturing skill and financial organizing ability? Who kept alive the practice of using the reheat cycle at a time in the late twenties and thirties when it seemed to be out of favor in the utility industry?

One could think of many other questions that would show additional areas of the untold story of steam power engineering. Perhaps this is a project for the recently formed Society for the History of Technology or for a university research center, as the State University of Iowa did in the case of the "History of Hydraulics." Future generations of engineers will be the ultimate beneficiaries of the perspective that will result from the publication of such a history.

Fig. 1—The compactness of the PCC 100,000-lb per hr packaged boiler is dramatically pictured above where the unit is shown in transport through the streets of Waterbury, Conn., headed for the Scovill Manufacturing Co. power plant shown prominently in the background

Packaged Boiler Meets Process and Power Needs

With the advent of the packaged water tube boiler it was only a matter of time before its unit capacity ratings would grow to a size sufficient for full scale industrial service. Here is the "largest" ever built now supplying steam at well over 75,000 lb per hr and at the same time meeting the load swings on a 400,000-lb per hr demand.

By CHARLES K. STICKNEY*

Scovill Manufacturing Co.

SCOVILL Mfg. Co. of Waterbury, Conn., the country's oldest brass manufacturer, has long been a strong believer in adequate plant power. It has been generating power continuously since 1802 and through the years has displayed a readiness to employ the latest advances in the technology. For example, in 1904 the company installed its first steam turbine, a 500-kw vertical unit; by 1919 its power plant was operating at the then daring pressure of 250 psi; by 1929 a topping turbine had been added. It is not surprising, then, to find Scovill installing the highest capacity "packaged" boiler ever built—a 100,000-lb per hr design—in its plant this past year.

* Chief Engineer of Power Plant.

The Plant Need

Behind the Scovill decision on this Combustion Engineering PCC 120 unit is the recurring industrial plant need to keep control over the different rates of load growth for process steam as against steam for electric generation. Scovill's power tradition, broadly stated above, had led to acceptance over the years of ever larger steam generators to benefit from the operating economies inherent in these larger units. As a result the steam needs for both power and process were being supplied almost entirely by two large boilers, one a 250,000-lb per hr unit and the other a 150,000-lb per hr design, both oil or pulverized coal fired and operating at 660 psi, 750 F total steam temperature. Several 250-psi, stoker-fired boilers of many years standing still in working con-

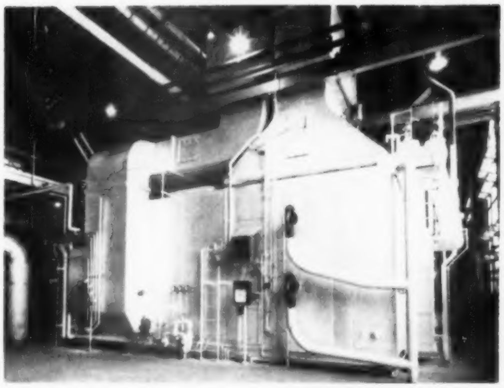


Fig. 2—Side view of the boiler in place shows fd ductwork on the left and the furnace gas outlet, above, and to the right

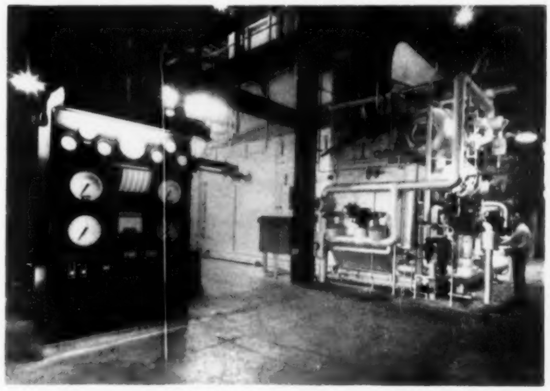


Fig. 3—Instrument panel and firing aisle of the packaged unit give good indication of the boiler's relative space requirements

dition served more for standby duty and only incidentally for process steam loads.

Over and above the problem of keeping ahead of the plant load growth was still another, a more pressing problem. This problem was the variable size of seasonal loads, the winter load demand on the one hand and the summer load demand on the other. The winter load, as of 1955, had grown to 400,000 lb per hr with every prospect of its reaching 450,000 lb per hr. This total load amounted to more than the then design rating of the installed steam generating capacity. Further, the summer load bid fair to reach 325,000 lb per hr. This latter condition made it imperative that the larger of the two operating boilers, the 250,000-lb per hr unit, be always in serviceable condition throughout the year and around the clock. In addition, the summer weekend load had climbed to 90,000 lb per hr. Obviously scheduled outages for annual boiler inspections or routine maintenance on the 250,000-lb per hr unit had to be limited to a weekend or else management would have to deliberately shut down much of the plant production.

The solution to this situation was clearly more steaming capacity and at a pressure level that would permit electric generation as well as process and auxiliary steam. Scovill began shopping for a boiler that would supply this steam need and yet be (1) flexible in loading range, that is, perform well over a fairly wide range of load as characterized by winter against summer needs; (2) relatively quick in response so that the boiler could absorb or assume the heavy demand swings created by various plant departments; (3) compact in physical dimensions so that the steam generator could be housed within the existing boiler house structure. The boiler, Figs. 1–4, was selected to fulfill these requirements.

The Boiler's Operation

This boiler, equipped for oil firing, has a design rating of 100,000 lb per hr at 660 psig, 750 F. The oil supplied to the unit is a domestic oil, relatively low in sulfur and comparatively free of vanadium. At the 100,000-lb per hr rating the oil reaches the burners at 150 psi and at 200 F. The unit is pressure fired with heated air for

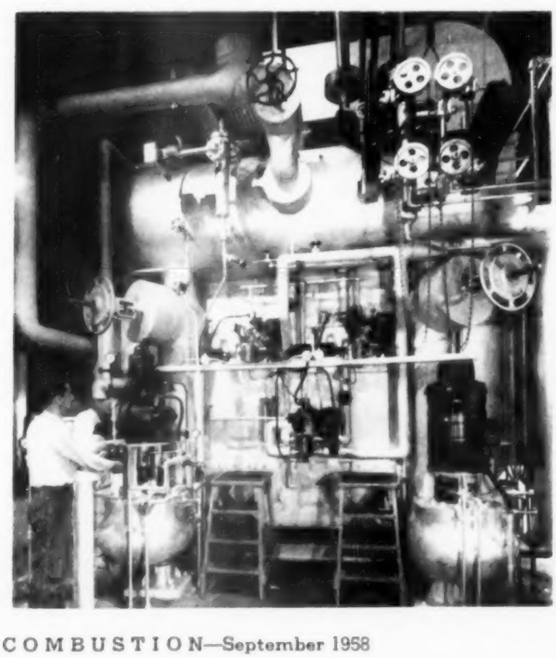
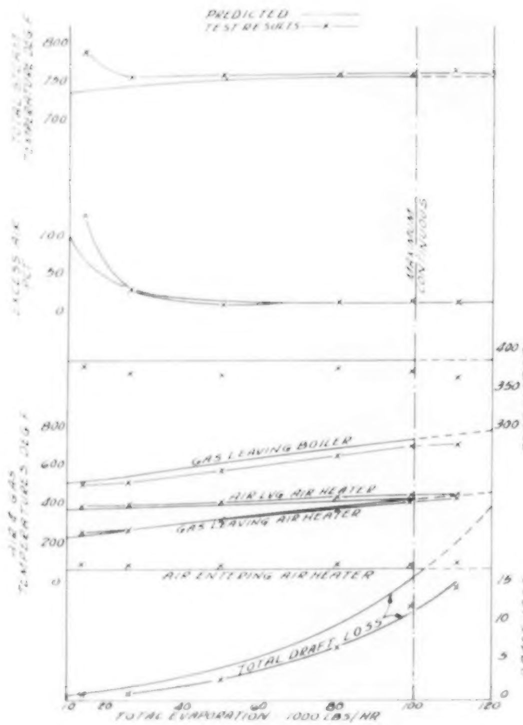


Fig. 4—Boiler front shows three burner assemblies; two boiler feed pumps, one on left a turbine-driven design and that on right a motor-driven one

TABLE I. PERFORMANCE DATA ON PCC BOILER

	Predicted Max Cont	Actual
Output, mm Btu/hr	101.65	102.83
Evaporation, lb/hr	100,000	99,000
Operating pressure, psig	660	660
TST, F	750	764
Feedwater temperature, F	390	376
Firing rate (18,500 Btu/lb), lb/hr	6,520	6,540
Excess air, pct	15.0	17.0
Release rates		
Per cu ft furnace volume, Btu/hr	93,000	95,800
Per sq ft EPRHS, Btu/hr	186,000	189,000
Gas temperature leaving boiler, F	750	718
Gas temperature leaving airheater (corr. for leakage), F	409	431
Entering air temperature, F	80	105
Air temperature at burner, F	440	467
Total draft loss, in. wg	16.1	12.1



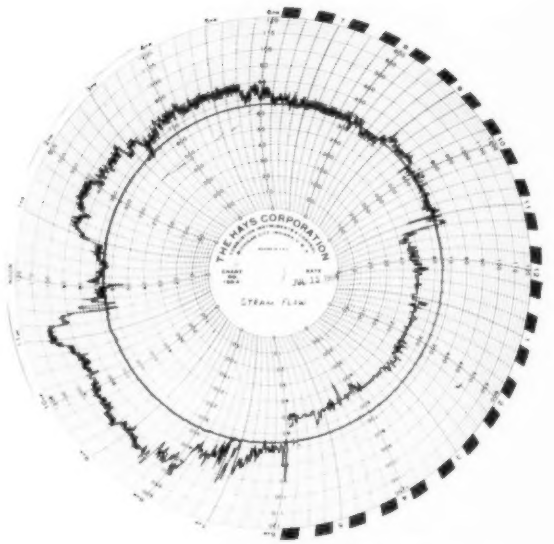


Fig. 6—Sample Monday steam flow chart pictures the load swings absorbed by the PCC unit as well as the time to get unit on the line

← Fig. 5—Plotted test results show salient features of the boiler's actual performance against predicted for full load range

combustion delivered by a separately supplied packaged Ljungstrom air heater. The full particulars for the boiler's operation make up the data in Table I, and the curves for Fig. 5.

All boilers are fed from a common water supply. The supply, taken from the Mad River, runs as high as 50 per cent makeup and requires external chemical treatment before admission to the boiler cycle. After treatment, a heat exchanger fed by drains from a continuous blowdown flash tank raises the temperature of the makeup water before it is admitted to two deaerating heaters. These deaerating heaters then supply the common feedwater main. Three pumps, two turbine-driven and one motor-driven, rated at 300,000 lb per hr each deliver the water to the boiler drums.

In the case of the new PCC unit a Copes flowmatic type A-O regulator is used. It is of the two-element type responding to rate of steam flow as well as to variations in boiler water level. The feed control is entirely independent of all other boiler controls and instruments. Should combustion control, flow meters or level recorders be out of service for any reason, the feedwater regulators continue on fully automatic operation, holding the boiler water level within close limits and providing the correct rate of feed input for the steaming conditions. This permits fast steaming boilers, which have comparatively little water storage capacity in their drums, to be operated at the optimum level of safety and efficiency.

Individual boiler combustion controls have been installed for each steaming boiler, but they work off a master controller so that a selected boiler regulator can be adjusted to absorb load swings. The PCC unit has been assigned this duty because it responds rapidly to these demands. It should be stressed at this point that the load variations in a plant such as Scovill's can be quite severe. The electrical demand registers sharp changes as billets, for example, are undergoing processing. An example of the typical load variations absorbed by the PCC boiler is shown in the flowmeter chart of Fig. 6.

Performance

In general the actual performance of the boiler equaled or exceeded the anticipated. Of particular note is the total steam temperature curve. The unusually flat characteristic of this curve is achieved without the necessity of steam temperature control equipment. The temperature is about 10 deg F higher than anticipated due primarily to low feedwater temperature. However, since all boilers deliver steam to a common header and the present boilers produce steam slightly under 750 F, the somewhat higher temperature from the newcomer is welcomed.

At the low end of the boiler's operating range a special condition peculiar to the Scovill plant exerts its influence upon practical loading limits for this PCC unit. The existing stack is relatively large and produces a high furnace draft with resultant high excess air and high steam temperature at loads below 25,000 lb per hr. Although it has been possible to improve the air flow control at low ratings by manual adjustment of the stack inlet damper, very infrequent operation at loads below 50,000 lb per hr make this adjustment unnecessary.

Fig. 5 shows quite well the boiler's performance over the complete range of load conditions. It is possible to bring the unit up to a rating of 120,000 lb of steam per hr and it was held at this level for one hour while steam purity tests were run. At the maximum continuous rating of 100,000 lb per hr results have been excellent.

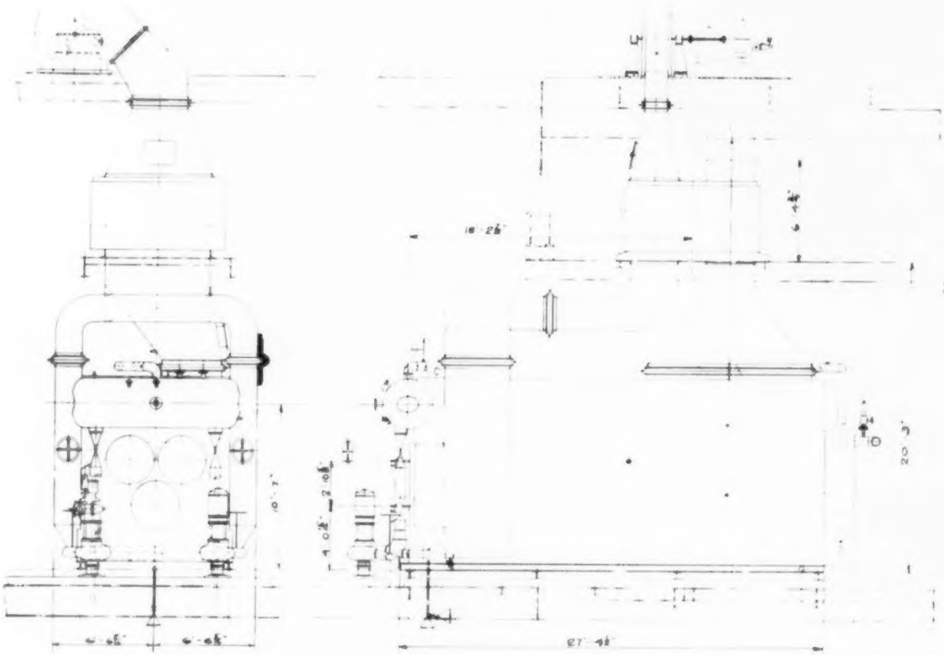


Fig. 7—Construction drawing gives interesting overall dimensions and locations of the boiler's major parts

and Table I compares actual against design for all pertinent factors.

Quick Response

The history of the controlled circulation boilers in marine and central station service led the PCC's designers to expect quick response to load changes as well as very fast startup and shutdown. This expectation has been met and the new boiler can be brought up to load in less than two hours. Similarly, it can be taken out for service in relatively short order.

This facility for quick startup and shutdown fits in very well with Scovill's need for tight schedules for maintenance and inspection.

Under usual load conditions the new boiler, as mentioned in the section on performance, serves to absorb the load swings and as such has lived up very well to the requirements for quick response.

Compact Size

The third essential the new boiler was expected to meet was that of relatively small quarters. The space allotted had been occupied by a 30,000-lb per hr unit and the most compact, modern, natural circulation design that could be placed in the same area was a 75,000 lb per hr unit. The PCC, rated at 100,000 lb per hr, comfortably fits into this section.

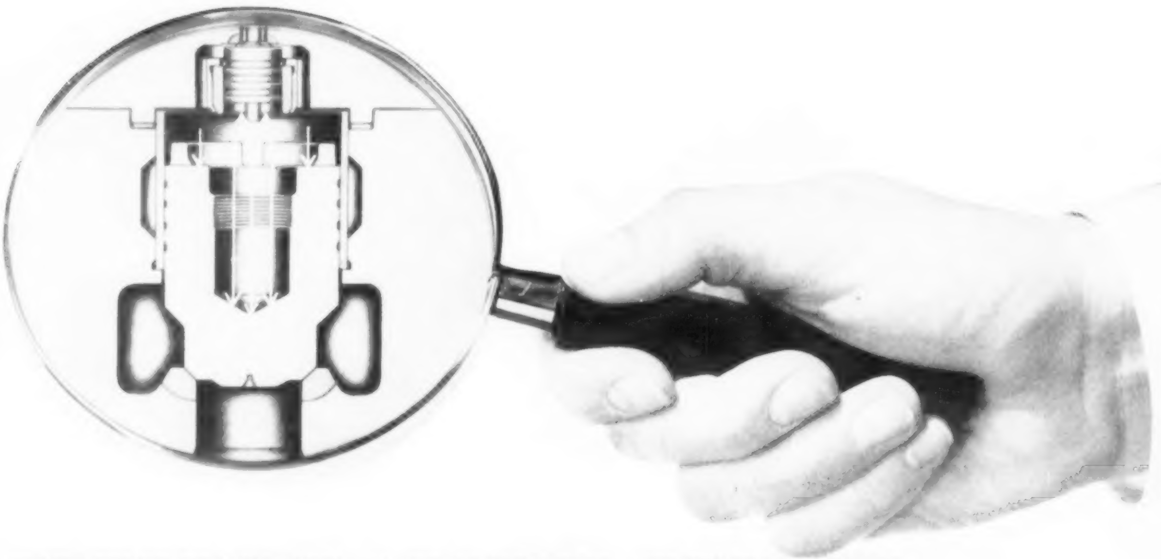
Fig. 7 gives the exact dimensions of the unit, but the boiler proper measures approximately 13 ft across the front by 34 ft deep by 13½ ft high. Existing building steel permitted installation of the Ljungstrom air preheater and the forced draft fan above the boiler.

Centrifugal Compressor for Soot Blower Duty

Electrical utility engineers will watch with interest the performance of an unusual, or what might be called an out-of-the-ordinary, type of air supply for boiler soot blowing and station service in a power plant. The new system has been selected for the 1,715,000-lb per hour Unit 8 in the Avon Station of the Cleveland Electric Illuminating Co., Cleveland, Ohio. Number 8 unit will be equipped with 138 Vulcan soot blowers using air as the primary blowing medium and controlled by a Selective-Sequence system.

The exceptionally interesting feature of the plant's air supply is that the air will be furnished by a fully automatically controlled centrifugal compressor, supplying air at 300 psi for the soot blowing and at reduced pressure of 125 psi for station service. The compressor is a 9160-rpm, 700-cfm, 2-cylinder machine with intercooling, driven by a 2500-hp motor.

The unique installation will provide the greatest flexibility possible in an air supply. The compressor may, on low air pressure and capacity requirements, furnish the plant with station service air at utilization pressure at low horsepower input. Then, when soot blowing time comes the operator merely changes the pressure set point. The compressor then swings into full pressure and capacity output to take care of soot blowing requirements. The required volume of air is available at all times and as long as it is needed, rather than having to interrupt the soot blowing sequence while air receivers are refilled.



By IGOR J. KARASSIK*
Worthington Corp.

The boiler feed pump and its associated equipment represent a major operating and maintenance consideration in today's power plant. Here in question and answer form is the second of a series of three clinic sessions on various boiler feed pump problems. The replies are the work of one of the topmost pump authorities and give specific information which we hope will prove valuable to our readers.

Steam Power Plant Clinic—Part II

QUESTION:

We are adding a 100,000-kw, 1800-psi throttle pressure unit to our system. The study of the boiler feeding requirements indicates that three boiler feed pumps should be installed, each designed for 450,000 lb hr against a discharge pressure of 2100 psi. Two of these pumps will operate in parallel at full load, the third pump being held for standby service. Because this latest addition will operate as a base load unit, our studies indicate that we cannot justify the use of a hydraulic coupling to operate the pumps at variable speed. Does this conclusion correspond to present practice?

ANSWER:

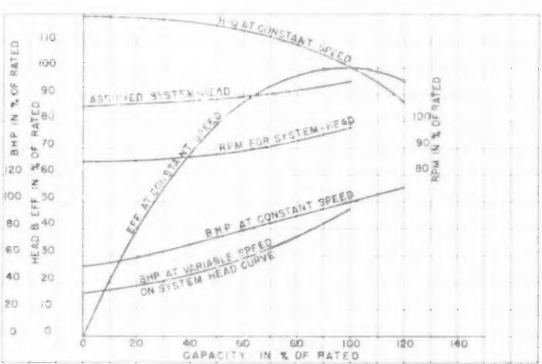
It is true that power savings from operation of a boiler feed pump at variable speed can more easily justify the additional expense of a hydraulic coupling when the load factor of the steam plant is such that frequent operation at part load is contemplated. To illustrate this, it may be well to consider a typical boiler feed pump performance curve, expressed in percentage terms of its rated design conditions. This curve shows the head-capacity of the pump at constant speed, an assumed system-head curve, the speed required to meet this system head curve and the power consumption of the pump both at constant and variable speed, Fig. 1.

It will be noted, for instance, that when the pump is operated at 50 per cent rated capacity, it needs to develop only 88.5 per cent of its rated head, and will run at 87.5 per cent of its rated speed. Its power consumption will be 52 per cent of rated power, compared to 74 per cent when the pump is operated at full speed.

Not all of the difference between constant speed and variable speed power consumption will be a net gain, as the variable speed mechanism has an efficiency which decreases proportionally with its output speed. Nevertheless, the savings in power consumption at part load operation are generally sufficient to justify the purchase of a variable speed transmission.

However, even when frequent operation at part load is contemplated, the choice between constant and variable speed operation tends to reflect personal preferences, because the justification of variable speed is intimately tied up with the principles followed in the evaluation of the savings. Within the past five or six years, about 25 to 30 per cent of the installations for 1250-psi throttle

* Assistant to Vice President and Consulting Engineer, Harrison Div.



TYPICAL PERFORMANCE CURVE
OF VARIABLE SPEED CENT. BOILER FEED PUMP

Fig. 1—From a typical performance curve a judgment can be made on advantage of hydraulic coupling.

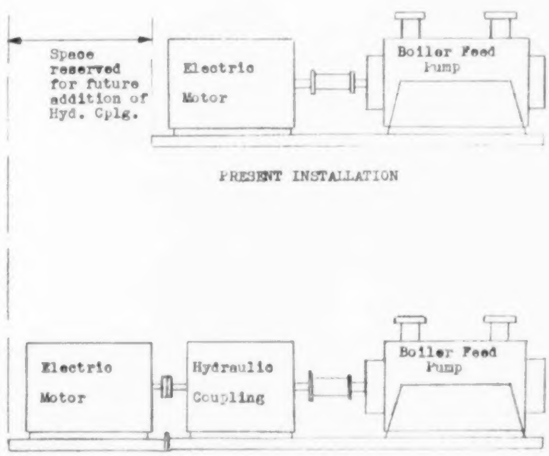


Fig. 2—Base loaded pump in five years may be handling swing loads and hydraulic coupling will help.

pressure and above have incorporated hydraulic couplings.

If the studies carried out do not justify the use of a hydraulic coupling, there is still a solution which would permit the steam power plant designer to "have his cake and eat it." A unit which is intended for base load operation now will probably change to a swinging load in another five or ten years as more efficient units are added to the system. If sufficient space is reserved beyond the electric motor, a baseplate extension can be added in the future, the motor moved back on this extension and a hydraulic coupling interposed between the motor and the pump. Fig. 2.

It is true that the pump speed will be reduced by 2 to 3 per cent because of the slip in the hydraulic coupling. On the other hand, the feedwater regulator can be removed, since speed variation will control flow to the boiler. The reduction in the required pressure to be generated will about balance the reduction of head due to the lower operating speed. The master control operating the feedwater regulator can be retained and used to control the hydraulic coupling.

In the particular case described here, a 1500-hp motor will be used. The length of a hydraulic coupling suitable for this size motor is approximately 6 ft. Thus, six feet of extra space must be provided beyond the motor itself for the future installation of the hydraulic coupling. At worst, these six feet of space may be wasted for a few years beyond the time when addition of the hydraulic coupling is predicted to be justifiable.

QUESTION:

Our power plant has two 15,000 kw units, operating at 450 psi. The two units operate independently, except for a tie-in at the boiler feed pumps, as shown on the sketch, Fig. 3. A motor driven boiler feed pump supplies all the feedwater required for each one of the units. A common standby boiler feed pump is arranged for turbine drive. This pump is started automatically on failure of either of the two motor driven pumps to develop sufficient pressure. Since it may have to take its suction from either one of the two deaerating heaters, its suction valves are kept closed. One of these is opened by the impulse which, after a suitable time delay, starts the turbine driven pump.

Obviously, the pump is not expected to be operated very frequently. But the very first time this pump started due to an emergency, about 6 months after its installation, the pump seized and was seriously damaged at all its internal running joints. What could have caused this failure and what can be done to avoid its repetition?

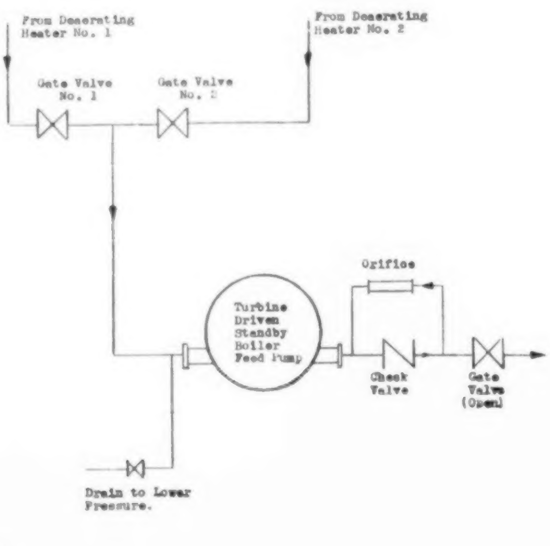
ANSWER.

The reason for installing two valves in the suction lines to the turbine driven pump is that the pressure in the two heaters and the temperature of the feedwater at the suction of the two individual pumps may be unequal if the two units are operating at different loads. Thus, the suction valves are normally closed and one of them (depending on which of the two motor driven pumps loses pressure) opens a fraction of a minute before the pump is put on the line.

While this arrangement satisfies the requirement that the two suction lines will never be interconnected, lest unequal pressures in the two deaerating heaters result in flashing and steam binding, it also acts to cause difficulties when the turbine driven pump is started up.

As it is stated in the question, this pump is apparently never operated under ordinary circumstances and is held only for emergencies. When such an emergency does not arise for six months after the pump has been installed, the pump is not in condition to be operated for the simple reason that leakage of water through the stuffing boxes will have drained the pump casing down to the level where no further leakage can take place and the pump is air-bound. This is undoubtedly what happened when the pump seized on start-up.

There are several means available to avoid the recurrence of this type of failure:



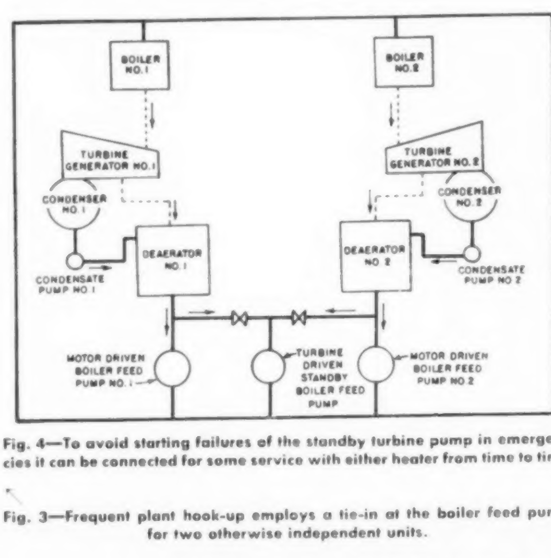


Fig. 4—To avoid starting failures of the standby turbine pump in emergencies it can be connected for some service with either heater from time to time

Fig. 3—Frequent plant hook-up employs a tie-in at the boiler feed pump for two otherwise independent units.

a) The pump could be operated manually at least once a week, say over the week-end. The suction valve to one or the other heater would be opened and a high point on the casing vented, just in case enough feedwater will have escaped at the stuffing boxes.

b) The pump can be always kept connected to one of the heaters. Thus, 50 per cent of the time it would be connected to the one unit for which it must replace the motor driven feed pump. In the event that it is the other pump that would have required replacement, the stand-by pump can operate temporarily taking its suction from the other heater. It can be switched into its proper unit manually, after the emergency will have been taken care of, Fig. 4.

c) A by-pass can be installed around the check valve in the discharge of the turbine driven pump, with an orifice in this by-pass. This orifice should be selected to pass a nominal flow of feedwater — say 3 to 5 gpm — under the pressure differential between the discharge header pressure and the pump internal pressure. To prevent building up pressure within the casing, a second by-pass or drain should be installed between the pump suction and some lower pressure reservoir, such as the surge tank for condensate into which the deaerating heaters overflow. In this manner, the pump casing will always be kept full of water and primed, ready to start. This arrangement will also accomplish the purpose of a warm-up line.

The by-pass around the check valve need not be closed after the pump has been started, since the flow through it will reverse and its presence will have no effect on the operation of the pump. The drain line from the suction to a lower pressure reservoir can be closed manually, at leisure, after the pump has been started.

Stoker Coke While Making Steam

What is believed to be the first car load shipment of stoker coke produced by a continuous process in conjunction with steam generation was made today by Island Creek Coal Company. Three carloads of the new product were produced on a test basis in cooperation with the P & L E Railroad at its plant in McKees Rocks, Pennsylvania and shipped to a chemical plant for use in producing calcium carbide.

This initial shipment is the culmination of more than two years of laboratory research on the production of stoker coke from Island Creek's Scarlet Flame low-ash coking coal.

In this process, coking coal is continuously fed to a traveling-grate stoker operated at approximately double its normal speed. By control of air admission, the volatile matter in the coal is burned to provide heat for coking and to raise steam, while most of the fixed carbon in the coal is recovered in the form of small-sized, porous, high-temperature coke. When exceptionally low-ash coal is used, the resultant coke is well suited for electrochemical and electrometallurgical processes such as the manufacture of calcium carbide, phosphorous, and ferro-alloys.


SECO

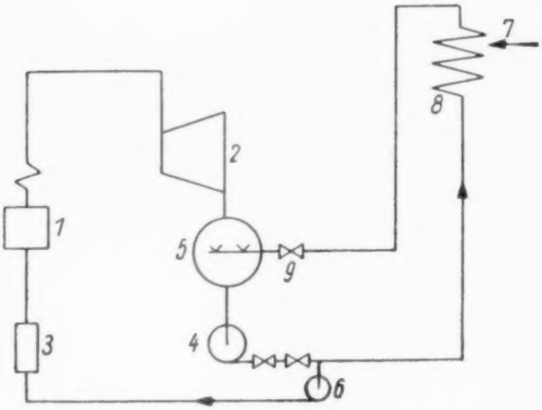


Fig. 1—Basic arrangement of Heller System of an air condenser

The following paper as explained in the footnotes is a translation of a translation and hence a real danger exists that the thought content may not be as accurate as desired. The idea of an air condenser, though, holds such strong appeal for all power men that we are publishing the material in the form in which we have it.

An Experimental Air Condenser In Hungary*

IN TODAY'S condensing power plants the turbine exhaust steam is condensed almost exclusively by surface condensers with water cooling. The availability of the required water quantities adds considerable difficulties to the selection of thermal plant sites. The location of large power plants is therefore influenced not only by the fuel supply, but by the water resources as well. The steam power plants are located at rivers and lakes, or at such places where there is a supply of makeup water as required for cooling towers or cooling ponds. The latter two cases need such makeup water because the condensing water is usually cooled by evaporation.

The idea to eliminate the requirement for condensing water for operation of condensing turbines by transferring the heat to be removed from the cycle directly to the atmosphere is by no means new. Air condensation of a type known as the GEA System has already been used in Germany. In this arrangement the turbine exhaust steam flows into a surface condenser which is situated next to the turbine room and is cooled by forced air flow. The exhaust steam is carried to the surface condenser under vacuum by a pipe with a diameter larger than the exhaust opening of the turbine. Because of these large pipe dimensions and operating difficulties due to leakages this system is not suited for larger power plant units. A 50 Mw turbine, for instance, would require two pipes with about 13 ft diameter.

The Heller Condensation System

The Heller System of air condensation uses a different principle and avoids the disadvantages of the GEA system. The basic arrangement is shown in Fig. 1. The surface condenser is replaced by a mixing or direct contact condenser (*5*). The mixture of condensing water and condensate is removed from the condenser through the condensing water pump (*4*). A portion of it is pumped as condensing water into the surface cooler (*8*) which is set up outside and cooled directly by air. The condensing water therefore circulates in a closed circuit filled by condensate. That portion of the mixture, however, which represents the quantity of condensed steam is pumped by the condensate pump (*4*) into the feedwater tank (*3*). Makeup water is not needed in this system.

A throttling valve (*9*) ahead of the mixing condenser keeps the system from the condensing water pump to the condenser under a small positive pressure. This insures that air cannot enter the cooling system through possible leaks. Larger installations might warrant replacing the throttling valve by a water turbine ahead of the condenser in order to utilize the pressure differential existing between the pressure of the cooling system and the condenser vacuum.

Because the Heller System of air condensation does not need makeup water, the problem of providing condensing water does not exist. This makes it possible to select the otherwise most advantageous plant site. Another advantage of the new system is the fact that the mixing condenser is cheaper and simpler than a surface condenser. It further eliminates the possibility of a temporary loss of turbine efficiency due to dirtiness of condenser tubing, as well as the equipment which would be otherwise needed for the treatment of condensing water.

Cooling the condensing water in this system requires heat exchangers with particular properties. Large heat quantities have to be removed with a small temperature differential available between the saturated steam temperature corresponding to the usual vacuum (95 F to 113 F), and the ambient air temperature. These conditions require a large cooling surface or a high air velocity. The possibility of using large surfaces is limited by the space requirements and first costs. High air velocities on the other hand, produce high air pressures which coupled with large quantities call for undesirably large fan power. A 10 Mw turbine already requires the handling of approximately 970,000 cfm for which the additional power to

* Excerpt from a report "A Légkondenzációs Kísérletek Eddigi Eredményei" published in *Magyar Energiagazdaság* or *Hungarian Power* No. 10, 1955, pp. 361–369 and translated and edited by Dipl Ing. E. Markus in *Energietechnik* Vol. 7, No. 10, 1957, pp. 448–453.

Translation from German to English by Dipl Ing. W. W. Schroedter, Combustion Engineering.

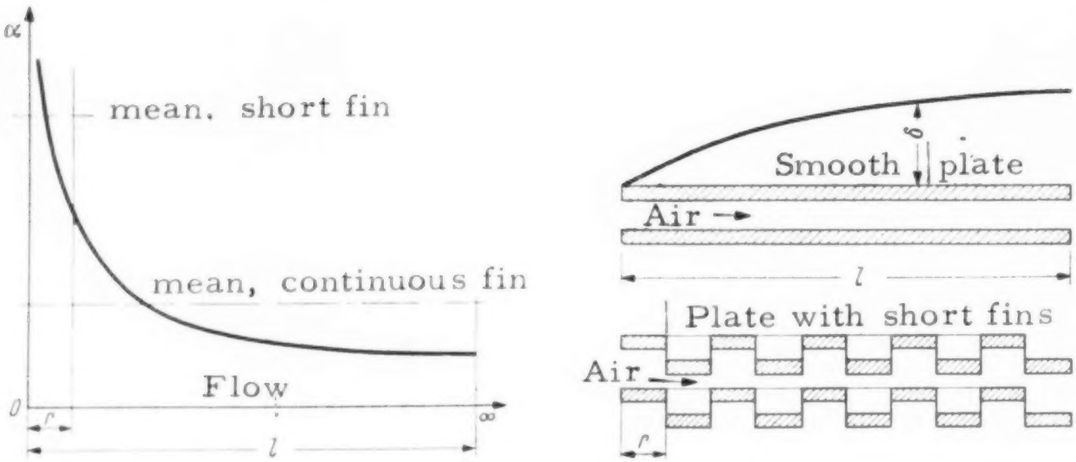


Fig. 2—Variation of film coefficient of heat tranfer shows how growth of a boundary layer on metal surface retards transfer

quirements would seriously affect the net station heat rate.

The Required Suitable Heat Exchangers

An economical air condenser plant therefore needs heat exchangers which will remove heat at low air resistance while being acceptable in weight, space requirements and first cost. The lack of suitable heat exchangers had indeed been the obstacle to the use of air condensation. The practical realization of the first plant of the Heller System was made possible by the development of such apparatus.

In order to gain practical experience in air condensation, a full scale experimental plant was set up. By revamping an existing condensation plant, first costs were held down while the minimum requirements for a full scale test were met. The steam turbine plant of a textile factory near Budapest with 830 kw output was used where the surface condenser was easily accessible and the installation of a mixing condenser did not present difficulties.

Design of Cooling Elements

New design principles were used to develop ribbed cooling elements (System, Beck-Forgo) which were specifically adapted to the special requirements of air condensation.

When gases or fluids flow along the surface of a solid body, the velocity of gas or fluid particles in direct contact with the surface will be retarded. These particles which are slowed down due to friction make up the boundary layer, extending along the body surface and increasing in thickness in the direction of flow (shown in Fig. 2).

If a temperature difference exists between the surface and the fluid, a heat transfer occurs in which the heat leaving the surface must cross the boundary layer in order to enter into the fluid flow. The retarded particles along the surface form an almost stationary layer with a temperature close to that of the surface. This layer of air which at the surface increases in richness in the direction of flow, prevents with increasing effectiveness the transfer of heat. The change of the film coefficient X along the surface in direction of flow is shown in Fig. 2. At $X = 0$ where the boundary layer has zero thickness the film coefficient X begins with an infinitely high value, but rapidly decreases as the boundary layer increases.

The principle applied now consists of preventing the formation of an isolating boundary layer in direction of flow by dividing the cooling ribs into very short sections. The shortened contact surface prevents a continuous boundary layer and thereby considerably increases the mean film coefficient, which reaches a value several times that achievable with continuous fins (Fig. 2).

The short rib system had already been developed by a radiator factory in Hungary and its aluminum radiator element was used as a design base for the heat exchanger of the air condensation plant. In adapting this element for the Heller System tests the existing manufacturing equipment had to be employed to facilitate fabrication. This facility with its avowed purpose of fabricating radiator elements, however, also put a certain amount of restraint on the Heller System component design.

A cooling element, as used for the air-cooled condenser at the textile plant, is shown in Fig. 3. It differs from the original radiator element basically only in its dimensions. Each element is $39^1/_2$ in. high, 79 in. long and 3.94 in. (100 mm) wide. It is made up of two main pipes of 2.36 in. O.D. and has 22 connecting cooling tubes. To each of the longitudinal fins on the cooling tubes are fastened four plates fitted with short ribs in a venetian blind arrangement. The plates are connected to the fins and the cooling tubes to the main pipes by pressing. The material used is pure aluminum which after fabrication was covered inside and out with a thin oxide film by a chemical process. The individual elements are connected to each other by loose trimmed flanges.

These elements used for the textile plant are not yet fully satisfactory because the thermal requirements for heating radiators and those for cooling elements in an air-cooled condenser do not completely coincide. A new element has therefore been since developed for which existing production facilities were disregarded and thermal principles only applied. This element consists of aluminum tubes fitted with aluminum plates carrying short ribs. This element is superior to the older model both

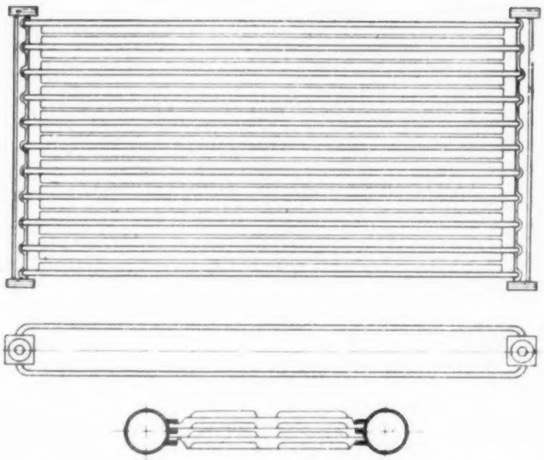


Fig. 3—Typical cooling element for air condenser has two main pipes (vertical in sketch above) with twenty-two connecting cooling tubes. Cooling tubes, lower figure in sketch, has venetian-blind type plates fastened on each longitudinal fin

technically and economically. Tests show an increase in heat transfer by about 80 per cent with the same material weight.

Description of Air Condenser

The air condenser was designed for condensation of 11,000 lb of steam per hr corresponding to a cooling capacity of 11,000 lb per hr × 940 Btu per lb = 10.35×10^6 Btu per hr. This would therefore rate the experimental unit for a 1-Mw turbine having a steam rate of 11 lb per kwhr. The existing turbine has 0.83 Mw output with 170 psig and 608 F at the throttle. A surface condenser was used and the condensing water cooled in a cooling tower.

The flow diagram of the air condensation plant is shown in Fig. 4. From the mixing condenser (*2*), which is arranged below the turbine (*1*), the condensate and the condensing water flow to the circulating pump (*3*). The pump is situated in a 13-ft shaft in order to obtain the necessary head since the condenser is under vacuum.

The circulating pump (*3*) pumps the condensing water into the cooling elements contained in the cooling tower (*4*) which is situated about 260 ft from the turbine room. The cooled water returns from the cooling towers through the pressure reducing valve (*5*) to the spray pipe of the condenser. The valve prevents the condenser vacuum from extending into the cooling elements and is set to maintain from 2.8 to 4.2 psig at the highest point in the cooling elements.

During startup, air is removed from the condensers by the high-capacity steam ejector (*6*). The three-stage condensing high-vacuum steam ejector (*7*) is used during normal operation.

The amount of condensate, which corresponds to the steam being continuously condensed, is diverted from the otherwise closed condensing water circuit downstream of the circulating pump (*3*). This condensate is handled by a separate condensate pump (*8*) which may be bypassed through piping (*9*). The circulating pump (*3*) moves the condensate on through the condensers of the high-vacuum steam ejector (*7*) and the water level regulator (*10*) into the condensate tank (*11*). It should be added that all condensate flow passes the condensers of the steam ejector (*7*).

The water level regulator maintains the required water level in the mixing condenser (*2*). If the level is too low, the regulating valve opens. The feedwater coming from the condensers of the steam ejector (*7*) will then flow into the mixing condenser instead of into the condensate tank (*11*) because of the higher pressure differential. The regulating valve closes and the condensate moves into the condensate tank (*11*).

The water level regulator also insures sufficient water flow through the condensers of the steam ejector at low load and no load conditions. In the event of a regulator failure the water level may be adjusted manually by using the bypass line (*12*). The system is initially filled through the connection (*13*) from the water treating plant. About 5300 gal of treated water are required.

In order to employ the cooling tower the water is piped to the storage tank (*14*) having a capacity of about 4000 gal. This tank is connected to the outlet pipe of the condenser by the line (*15*) and is arranged as far as possible on the same level as the mixing condenser. This makes it possible to store the water drained from the cooling tower both in the mixing condenser and in the

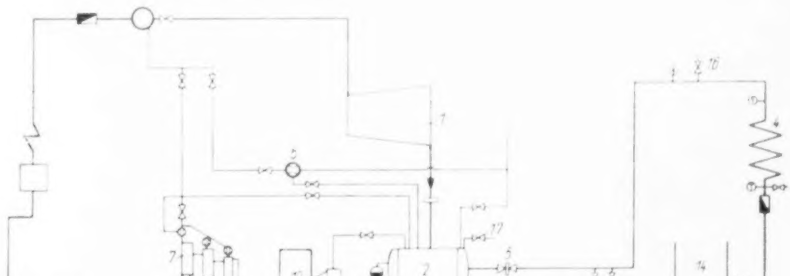


Fig. 4—Flow diagram of air condensation plant

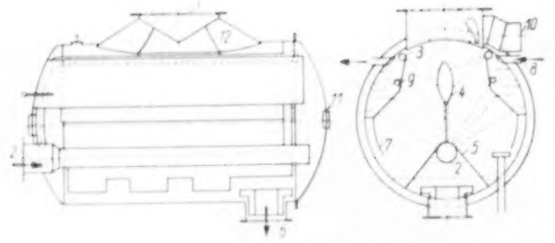


Fig. 5—Mixing condenser is arranged as shown

storage tank. When starting up during cold weather it may be necessary to preheat the stored water with injected steam.

The mixing condenser is to condense 11,000 lb of steam per hour and to transfer 10.35×10^6 Btu per hr to the condensing water which is to have a temperature rise of 14.4 deg F. The differential between the saturation temperature of the exhaust steam and the condensing water at the outlet should be less than 1.8 deg F. The required condensing water flow is 1410 gal. per min.

The mixing condenser has a shell with 5.35 ft diameter, 7.87 ft long, Fig. 5. Steam enters the condenser through the upper opening (*1*). Condensing water enters through opening (*2*) into the main spray pipe containing 1252 nozzles of 0.118 in. diameter in eight rows. Two other spray pipes (*3*) which contain 660 nozzles arranged in two rows on each pipe spray their water against a foil (*4*). The mixture of steam condensate and condensing water collects below the screen plates (*5*) which support the main spray pipe. The water leaves through opening (*6*) to the circulating pump. On both sides of the condenser, baffles (*7*) form a separated space which opens to the condenser at the lower end and from which the air is removed through the pipes (*8*). The cascades arranged in these spaces are to help condense any steam flowing upwards to the pipes (*8*) by having water from the spray pipes (*9*) cascading down in counterflow. This will leave practically only air to be handled by the steam ejector. The lower ends of the baffles extend to about 8 in. above the normal water level. It is from here that the collected air is removed. A connection (*10*) at the steam inlet for the safety valve and observation openings (*11*) at both ends of the condenser are provided. The steam inlet connection also has vanes (*12*) installed to distribute the steam flow.

Arrangement of Cooling Tower

The cooling elements are arranged around the perimeter of a circular concrete base of about 42.5 ft with two elements each in a V arrangement below a roof as shown in Fig. 6. The center is taken up by the aerodynamically shaped air inlet cone, topped by a 49 ft diffuser which is to regain a part of the dynamic energy of the exhaust air.

Space is provided below the base for piping and valves. The condensing water is piped to a center distributor and flows from there into the branches at the perimeter and into the cooling elements.

The condensing header first flows horizontally through element I, Fig. 6, then vertically through the element header into the element II. It continues again horizontally through element II and then upward into element III and so forth through element IV. The upward flow allows venting during startup. The cooled condensing water returns from the uppermost element by way of the ring header on the roof and connecting pipes into the condenser. Each of the 27 cooling columns had eight elements and because of the experimental character of the equipment is fitted with an inlet and outlet shutoff valve.

In entering the fan the air passes the extended surface of the 216 cooling elements which are offset to each other to avoid air bypassing. The parts without extended surface are sealed by plate against infiltration. The elements are held at one end only to permit expansion.

The fan rotates on a vertical axis with four profiled blades at a pitch angle which may be changed when the fan is stopped. The fan whose outside diameter is 19.7 ft has a belt drive with bevel gearing from a 25 kw motor. The highest revolution is 130 per min which may be reduced through a series resistance in 12 steps.

Operation

The experimental air condenser was ready for operation at the beginning of 1954. It was soon found that the water side pressure drop in the condenser was larger than anticipated. This was due to the drop across the spray nozzles which had been designed for the customary flow coefficient of 0.5. It was found that for nozzles arranged in rows and in comparatively thick piping, as used here, a coefficient of 0.4 should have been used. After increasing the bore, the available head could establish the design circulation of 1410 gal per min.

The equipment was at first operated in one shift. Although this puts the equipment through a more severe operation it is less suitable for testing than continuous load. It was noted that the oxygen content of the condensing water increased during the shutdown periods. This increases the danger of corrosion despite a reduction of the O_2 content after startup. Beginning of 1955 the plant was operated continuously and has demonstrated its usefulness.

The cooling tower was designed to establish the same operating conditions for the turbine as would be achieved with a standard evaporative cooling tower. The characteristics of air condensation, however, establishes this cooling capacity at a certain air temperature only. Below this temperature air condensation will show a larger and above this temperature a lower capacity.

The usual operating conditions for a tower with evaporative cooling are: at 59 F and 70 per cent relative humidity of the ambient air the temperature of the cooling water is to be reduced from 98.5 F to 80.5 F. A 9 deg F differential establishes a condenser temperature of 107.5 F.

The change in condenser temperature in relation to ambient air temperature for a standard cooling tower designed for above conditions is shown in Fig. 7. The graph also shows the condenser temperatures with air condensation.

The cooling capacity of the elements is determined by the design of the heat transfer surfaces, as well as by the temperature and flow rate of the air and the water. In selecting these design conditions, the capacity of the available condensing water pump and other factors had to be considered.

The air temperature at which both cooling systems

would produce the same condenser temperature is 56.5 F when the air condensation system is designed to have a condenser temperature of 109.5 F with an ambient temperature of 59 F. In the air condensation system the condenser temperature varies linearly with the ambient temperature. At 41 F ambient the condenser temperature will drop to 91.5 F. Since this is the optimum value of vacuum for the turbine, there is no need to utilize any lower air temperature. When compared with the standard cooling tower, air condensation will, as shown in Fig. 7, produce a higher vacuum when the ambient is below 56.5 F and a lower vacuum when the ambient is higher.

It may be noted here that due to the meteorological conditions around Budapest the air condensation system will improve the standard condensation with cooling tower during 220 days and will be less efficient on 145 days.

It was considered important to determine the actual relationship between the ambient air temperature and the temperature of the condensing water entering the cooling elements. For it is a characteristic of the mixing condenser to have a saturation pressure corresponding to the temperature of the condensing water leaving the condenser.

The cooling system was designed to condense 11,000 lb of steam per hour with 1410 gal of condensing water per min, top fan speed and all cooling elements in operation. When the tests could not be run in this manner, the results were adjusted for the design conditions. The measured average condenser temperatures are plotted in Fig. 7 against the ambient air temperature in curve *c* which exceeds the curve *b* representing the design conditions.

The temperature differential between the condenser and the ambient air was calculated at 50.5 F. The actual differential, measured at 59 F, is due in part to production deficiencies of the cooling elements and in part to an air flow lower than calculated. The latter was later eliminated by changing the pitch angle of the fan blades.

The temperature differential was reduced to 54 F but the fan power increased somewhat beyond the predicted value of about 1.4 kw per 1000 lb per hr steam.

This comparatively low value is due to the low air velocities which the ribbed cooling elements utilize to their advantage with a resultant low air resistance. The heat transfer may be reduced through bypass air and dirtiness. Leakages were easily found and removed during initial operation. Dirtiness of cooling elements by air dust was only slight after one and a half years of operation despite strong dust formation in the vicinity of the cooling tower. The dirtiness could be removed during operation by simple washing of elements.

It was found that the predicted heat removal of 10.35 $\times$ 10^6 Btu per hr could be released with an air flow of 360,000 to 380,000 cfm, an air temperature rise of 25 to 31 deg and a cooling differential of 14.5 deg F. The heat removal of the cooling tower may, of course, be increased by higher fan output. Windy weather also increases heat removal.

The cooling elements did not perform as predicted only where production deficiencies had caused insufficient metallic connection between ribs and tubes. As mentioned earlier, an improved design has in the meantime been perfected.

The fully welded piping connections between the condenser and the cooling tower, as well as the rolled cooling tube connections, prevent any water loss. The cooling elements were tested prior to erection with 71 psig.

The cooling tower was operated during winter without serious inconvenience. The sudden heating during refilling with preheated condensing water also did not present any difficulties. During winter it is important to drain the water as quickly as possible when shutdowns occur. The horizontal arrangement of the cooling elements makes this procedure somewhat time consuming although automatic air suction waves (*16*, Fig. 4), have been provided. During filling of the elements they function as vent valves. The newly developed cooling elements now have a vertical tube arrangement.

Winter operation, however has shown that this cooling system remains operative even at very low temperatures and that there is no possibility of ice formulation as occurs on standard cooling towers. Summer operation, too, was so satisfactory that there should be no reservations about building larger cooling systems.

Attention was paid to the mixing condenser to determine the effectiveness of the cooling water spray which is to condense the steam by heat removal. Ideal heat transfer in the condenser would allow zero temperature differential between the entering steam and the leaving condensing water because each water drop has had suf-

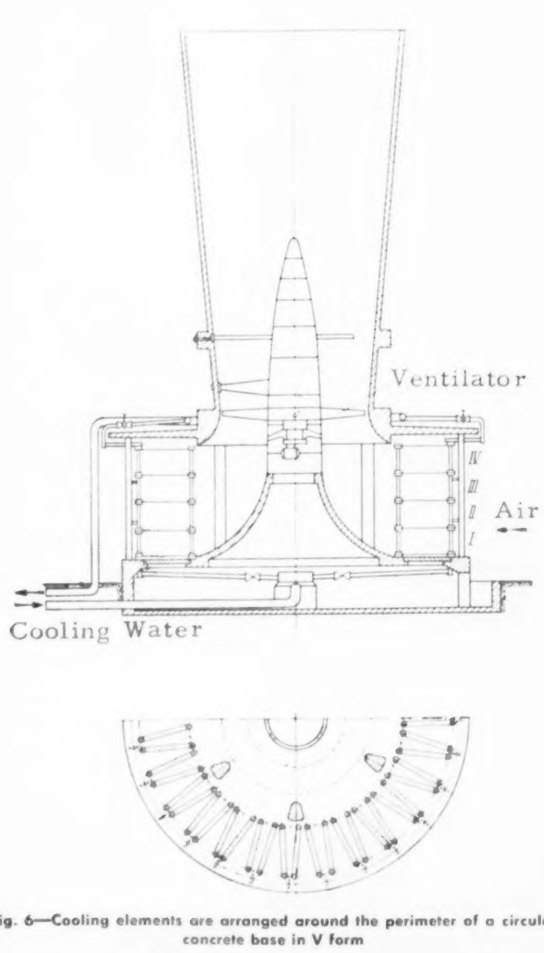


Fig. 6—Cooling elements are arranged around the perimeter of a circular concrete base in V form

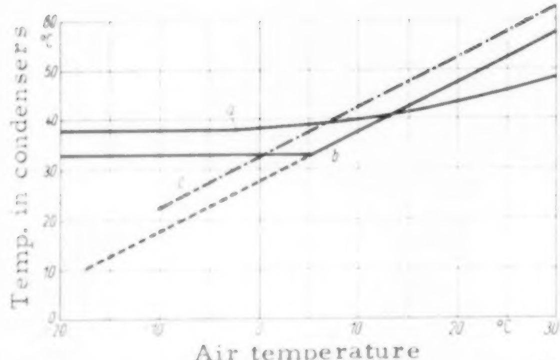


Bild 9. a üblicher Kühlturm, b entworfene Luftkondensation, c ausgeführte Luftkondensation

Fig. 7—Change in condenser temperature with that of the ambient air temperature is portrayed above

ficient contact with the steam. Good atomization produces a large water surface and therefore effects good heat transfer.

Actually the temperature differential amounted to less than 1.8 deg F and was often below measuring accuracy. The differential also remained at only 1.8 deg F when the steam load of the condenser exceeded the design value by 20 per cent. When not restricted to the dimensions of the former surface condenser a new design could considerably reduce the rise of the mixing condenser, especially if proper spray nozzles will improve atomization of the condensing water.

The power required for recirculation of the condensing water is about 2.8 Kw per 1000 lb of steam per hr with a pump efficiency of 77 per cent. This brings the total auxiliary power at 11,000 lb of steam output per hr to 4.2 kw per 1000 lb of steam per hr.

These data were found during 2000 hours of operation. During that time no incidents occurred which might indicate any basic deficiencies.

Only in starting up and shutting down does operation present novel aspects. During winter the condensing water temperature should be more carefully controlled. To avoid excessive cooling at very cold weather, the fan speed has to be reduced or the fan has to be shut down altogether. Some cooling elements may also be cut out. During special conditions such as minimum load or startup, the condensing water can be heated by steam injection.

Winter operation presented certain difficulties only when the generator or the condensing water pump tripped out suddenly. In order to accelerate the shutdown operation, due to a pump failure, an automatic valve, *17*, Fig. 4, was provided at the condenser to immediately break the vacuum. This valve is actuated when the power circuit of the pump is broken.

For rapid drainage of water from the cooling tower the air suction valves provide a sufficiently large air opening. The shutdown operation was later simplified by providing a remote control of the drain valves *18* and *19*, Fig. 4, from the turbine control board. Because both valves are mechanically interconnected, the formerly somewhat complicated shutdown operation is now reduced to one manual operation. Everything else follows automatically. In other respects, too, operation demands nothing unusual from the operating personnel.

Further long term tests are to show whether corrosion occurs in the condensing water circuit, consisting of aluminum and iron sections, since it is known that strong alkaline water (pH > 8) will attack aluminum and acid water will attack iron. The condition of the condensing water, as well as possible corrosion of the cooling system, should therefore be regularly checked.

Conclusion

In conclusion, it may be stated that the air condensation of the Heller system has lived up to expectations. The system is suited for larger power plants also with regard to the novel cooling tower, the mixing condenser and reliability of operation. For areas with low water availability this system may be considered the most suitable solution. Overall investment is lower despite a higher cost for the new type of cooling towers. It should be noted that the new system eliminates investment and operating expenses which normally would be incurred for procuring and treating of the condensing water. It can therefore be assumed that the new system of air condensation can compete with the old type cooling tower even at locations which are not handicapped with regard to availability of condensing water.

Paris Conference on Combustion of Solid and Pulverised Fuels—Dec. 1957[*]

G. G. THURLOW

THIS Conference was concerned with the whole problem of the combustion of solid and pulverised fuels. It was organised by l'Institut Français des Combustibles et de l'Énergie and, in all, over 40 papers were presented to an audience which was mostly French though there was a fairly strong contingent from Germany present together with representatives from Great Britain, Belgium, Yugoslavia and other countries. There was a short period for discussion after each paper but, chiefly owing to lack of time, this was not exhaustive.

(1) Coal Preparation

One of the first papers to the Conference discussed the future needs of French industry, mainly in relation to the position in 1956; and it would be interesting to compare this with the equivalent British requirements as outlined, for example, at the recent Institute of Fuel Conference on pulverised fuel. Of the 79 million tonnes† of solid fuel used in France in 1956, about 47 million was consumed as coal, the rest being converted to 19.3 million tonnes of coke and 8.8 million tonnes of briquetted fuels. About 14 million tonnes of the coal burned directly was used in the generation of electricity and about the same amount in industry in general, excluding steelmaking. About 10 million tonnes was burnt on domestic or other small appliances, together with most of the briquetted fuel. Nearly 70 per cent of the coke manufactured was used in steelmaking. Of the total fuel used, over 18 million tonnes had to be imported, about half of this being used for coking purposes. The characteristics of the fuels required by manufacturers and users of plant were outlined in another paper, particular emphasis being given to the possibility of using low-grade products.

The problems associated with grading, drying and washing of coals were discussed, probably the most interesting point being the development of a theory for washing in which the yield from any washing process is related to the density of the material. A coefficient is inserted dependent solely on the method of washing used, and values for this are given for various plants.

(2) Characteristics of Coals Used in Combustion

The second part of the Conference was concerned with tests for determining the various characteristics of coal of interest in combustion processes. In particular the relevance of laboratory tests to the selection of coals for use in domestic appliances and small boilers was considered. The measurement of volatile matter and the study of plastic properties of the fuel were considered. The effect of mineral matter was discussed and the practical value of a laboratory test in relation to the actual behaviour of the fuel in a furnace under different temperature and atmospheric conditions was dealt with in some detail. Such matters as the relevance of laboratory determinations of the softening temperature of ash were considered and comparisons drawn between the standard methods proposed by various organisations. A further paper also dealt with the fusibility of ash together with another important fuel characteristic, namely grindability. This discussion was occupied principally with the relevance of the use of the Hardgrove Index as a practical measure of the grindability of coal (see also Section 4.1).

(3) The Mechanism of the Combustion of Solid and Pulverised Coal

The Conference next considered the mechanism of combustion of coal and this section was opened by a valuable survey paper on the kinetics of the combustion of carbon, dealing with the reactions between carbon and oxygen, the influence of inhibitors, the formation of CO and the effect of gaseous diffusion in the combustion of a solid suspended in air. It was followed by a paper from Great Britain on the mechanism of the combustion of pulverised fuel. This again was basically a survey considering the combustion of pulverised fuel in terms of aerodynamics, thermodynamics, kinetics and thermophysics. The paper served to reveal the large number of unknowns in this subject and, whereas one can describe quantitatively the kind of reactions that take place, there is no doubt that a full description of these processes is

* Review No. 175. British Coal Utilisation Research Assn. Monthly Bulletin, Vol. XXII, No. 4, April 1958, Randalls Road, Leatherhead, Surrey. Printed by special permission.

† 1 tonne = 0.9842 British tons = 1000 kilograms.

exceedingly difficult. The point was made that it is essential to consider a flame as a whole and in fact the problem can only really be dealt with statistically. The differences between the volatile and coke-burning parts of the fuel, the effect of particle size on the rate of combustion, the determination of the relevant surface areas of the particles, the influence of the internal surface area, the formation of cenospheres and so on were all raised as subjects needing further thought. As the beginning of an answer to this series of questions, a report was given on the early work on the pulverised-fuel-fired furnace of the International Flame Research Organisation at Ijmuiden in Holland. The furnace was described and the results of the first trial on it were outlined. It was pointed out in discussion that the size of the Ijmuiden furnace was still small compared with many large furnaces, though, even here, the peak emissivity of the flame was approaching unity. Interesting results had already been obtained regarding such matters as the relative rates of combustion of volatiles and coke, the effect of the primary air quantity on the flame radiation, and the influence of secondary air velocity, and it was obvious that there would be much interest in the results of future trials.

This section was closed by two papers on the calculation of the heat transfer in pulverised-fuel-fired water-tube boilers. Firstly, the effect of direct radiation, re-radiation and diffuse radiation on the temperature of the wall screens was considered. A method was given for calculating the local temperature of these tubes in the region facing the flame. Allowances were made for tube thickness and the effect of slag layer (the effect of the latter appeared surprisingly small to the reviewer). The second paper gave nomograms for calculating the mean furnace temperature based on the theoretical furnace temperature and fairly standard heat transfer equations.

(4) Combustion Plant

With the beginning of the fourth part of the Conference the subject changed to combustion plant, and this section formed the major part of the Conference. Firstly, the firing of pulverised fuel was considered.

(4.1) *Firing of Pulverised Coal*

Two papers dealt with the transport of fuel. The hydraulic transport of sludge described in one of these is rather a specialised problem, but the paper on the transport of pulverised fuel in air is of wide interest and was particularly valuable in illustrating some of the difficulties arising owing to, for example, the differences in flow path of the particles due to centrifugal force, and separation of the coal-air mixture in mills. It was stressed that this paper was a presentation of empirical facts and that there was a need for a theoretical treatment of this important subject. Mention was made of the difficulties of developing such a theory owing to the lack of any homogeneous distribution of the coal particles across a pipe. The splitting of a coal-air mixture into two or three equal streams was also studied. A paper on drying was specifically concerned with the practical problems posed by the use in Belgium of low-grade fuels. It was stated that pre-drying was necessary for fuels containing more than 8 per cent moisture. When the Conference turned to consider mills it was stressed that these had other functions besides grinding. Of these functions, drying is obviously one, though there are other factors such as the need for an air/fuel ratio which must in general be kept constant at all fuel loads. It is not possible to consider scaling up the linear dimensions of present mills indefinitely. The requirements of the various components to cope with an increased throughput vary. Different types of mills were discussed and the effect of increase in throughput on the design of the various components was outlined. For example, for the crusher type of mill, the size of the mill itself will become smaller relative to the separator. Some practical aspects of mill operation were discussed including such problems as the removal of metal bodies before the fuel enters a mill. The difficulties in the study of the size distribution of pulverised fuel from a mill were mentioned and BAHCO type of sizing apparatus for small particles ($<74\mu$) was described. Mention was made of the causes and avoidance of accidents in mills and the ASTM method of comparing mills was outlined. The Hardgrove method of determining grindability was mentioned again and its usefulness was supported in so far as it was a direct model of an industrial plant.

The Conference then turned to discuss the burning of pulverised fuel, first of all in the more conventional types of chambers. It was immediately involved in the now somewhat familiar argument on the relative merits of the dry-bottom and slag-tap furnaces. It was stated that in firing dry-bottom furnaces it is preferable in practice to keep the primary air low with coals of less than 17 per cent volatiles in order to obtain rapid heating of the fuel entering the combustion chamber. A higher primary air rate is preferable with richer coals. With the dry-bottom furnaces, however, the disposal of ash was a major anxiety and, in fact, this theme recurred repeatedly throughout the Conference. The disposal of the fly-ash is obviously particularly serious with high-ash coals, and this was emphasised in some cases quoted. The use of this ash in cement-making was raised here and again later in the Conference but, though it is clearly possible to use it for this purpose, it was stressed that the cement produced could not have the uniform properties of normally-produced cement and would, therefore, probably only be of use in large constructional work where it could be used more or less as a filler. The amount of ash that can be tolerated in a dry-bottom p.f. furnace was discussed. In one station in Spain fuel containing more than 31 per cent could not be burnt successfully. It was suggested, however, that no general value could be given as it depended very much on the composition of the volatile matter from the fuel. For example, if there were plenty of hydrogen present this value could be increased to 40 per cent without trouble. The burning of lignite in Western Germany in dry-bottom furnaces was described in some detail.

An account was then given of the slag-tap furnace, and this included a report on some theoretical work and combustion performance data on this type of furnace. Recent German installations were described—particularly a "U" type furnace with roof burners, with temperature data for the gases at various points through the furnace.

Papers from Germany were then presented on both the vertical and the horizontal cyclone furnace. The vertical cyclone was strongly advocated, design considerations and actual plant details being presented, but the present bias toward the horizontal cyclone was empha-

sised by a table giving installation details of 44 horizontal cyclone furnaces installed in Europe. A paper by Gumz, identical with that presented to the Institute of Fuel Conference in London on pulverised fuel, summarised the rather unsettled opinion on these furnaces in Germany at the present time.

The final paper in this section discussed the firing of rotary cement kilns. The process was described and the methods by which the heat transfer in this type of furnace could be varied, i.e., by altering air velocity, rate of rotation of furnace, etc., were outlined. It was stated that most of the heat was transferred by radiation and not convection and some experimental data were given on an actual kiln. Whereas the experimental results indicated that the radiation from the flame decreased with increasing primary air (a result which agrees with the work in Ijmuiden), the output from the furnace in fact increased. It was suggested in a discussion that this paradox could be explained by the action of convection "vive," but there are insufficient data to provide an explanation at this stage. (The Reviewer was struck by the similarity of this result to the "steam oil" ratio controversy between Ijmuiden data and experience on the open hearth.)

(4.2) *Large and Medium Sized Installations (Excluding P.F.)*

The Conference turned next to the consideration of methods of combustion, other than with pulverised fuel, for large and medium sized installations. The first stoker considered consisted of a conventional travelling grate with, however, the pneumatic injection of part of the fuel over the top. The proportion of fuel that is burnt on the grate and the proportion that is injected over the grate could be varied, the amount burnt on the grate varying between 0 and 60 per cent. Two different fuels could be used and this suggests that this method of firing may be a means for burning fuels with a high proportion of fines. The advantages claimed for this process include an increase in the loading of the grate per sq ft, more uniform combustion rate along the grate, and an increase in flexibility, especially as the output can be controlled on only one half of the fuel fired. This means that one can get a quick response to load changes with consequent advantages in fully automatic control. The volatile matter of a fuel at present used on this grate in France is 20–25 per cent, but it should be possible to use coals down to 16 per cent volatile matter. The burning of coals with ash contents of up to 40 per cent should also be possible. Grate loadings of 40–45 lb sq ft-h are possible compared with the usual 30 lb sq ft-h.

Another process described was that using the "Ignifluid" grate, where the combustion takes place in a fully-fluidised bed, maintained by the primary air jet. The ashes settle at the bottom and are removed by a narrow sloping travelling grate. The sloping sides of the chamber containing the fuel bed are protected by a layer of coal particles. The first installation burnt anthracite fines, but crude slurries with a high percentage of water can (as was shown by a film) also be burnt direct. It is claimed that this process produces very little smoke.

In the "Stouff" method of combustion described next, a fluidised bed is also used but with a lower particle density. The combustion takes place in a cone-shaped chamber with an air entry in the form of a large slot at the bottom. The ash conglomerates and falls downwards and the whole process is basically very simple. The burning zone is a function of the air velocity and in the earlier processes this led to a lack of flexibility which has been improved by new techniques. It was suggested that this might be a possible carbonisation process.

The last of the combustion methods described was that developed by Lemaire, initially for firing anthracite fines. The combustion takes place on a mechanical grate installed in a very small primary furnace with a refractory roof extending over about a third of the length. The combustion products flow from this primary chamber into a larger chamber with a carefully-induced swirl caused by the furnace shape and the position of the secondary nozzle. Fines are re-injected.

(4.3) *Grates for Burning Special or Waste Fuels*

The Conference then turned to the consideration of the combustion of a variety of unusual fuels. These included bituminous shales, containing about 75 per cent of mineral matter, apple-stems and domestic waste.

It is to the great credit of the engineers involved that the efficiency of the pulverised-fuel plant burning the bituminous shales was claimed to be 82.1 per cent compared with 88.4 per cent for the same plant using coal.

The problems of burning the high-volatile fuels are, of course, different from those discussed up to now, as here a large part of the combustion is in the gas phase. Firing was mainly on to sloping grates. An interesting paper was concerned with the preparation and combustion of a particular lignite containing some fossilised wood and having a large water content. Various methods of burning this fuel both on grates and as pulverised fuel were described, probably the most novel being the combination of a p.f. unit with, at the bottom of the combustion chamber, a gas producer unit. Fuel dropping during the pulverised-fuel firing falls into the gas producer unit where it is gasified, the gases being burnt in the furnace proper. By this means, an overall efficiency of more than 85 per cent can be obtained.

The increased use of peat as a fuel in the U.S.S.R., Ireland, Yugolasvia, Scandinavia and elsewhere was stressed in one paper, which included a thorough world survey of the use of this particular fuel, together with a description of some of the plants used. Many of these were grate-fired, though it was stressed that more and more milled peat is being used in the more modern plant. Examples were given.

(4.4) *Domestic Heating*

The use of anthracites, cokes and semi-cokes within the size range of 30–120 mm (about 1–5 in.) on three central-heating plants was described, these being the French Nordon SD, the Dutch "Emma" and the German Strebel furnaces. The application of automatic control to this type of appliance was dealt with in a further paper. The necessity for developing automatic control (at a constant output) (*a*) to compete with oil and gas firing, (*b*) to cope with the increased use of low-quality fuels and fines, (*c*) to burn various fuels on one plant and (*d*) as an essential step in the development of larger central-heating plants, was stressed. Automatic fuel feeds and especially the use of screw feeders were discussed and various methods of ashing, chiefly semi-automatic, were described. The V.A.T.C. totally-automatic plant was discussed in some detail. In this, the coal is fed either by

gravity or by a piston and there is fully-automatic ash removal which is linked to the coal feed. A boiler burning anthracite was also described, in which the coal feed was controlled by a screw mechanism which lifted the coal continuously from the bunker to the top of the unit. This unit was again fully-automatic and in this case the cleaning piston was not only linked to the rate of feed but was also fitted with an over-ride which operated when the gas temperature below the grate fell below a set value.

In the discussion of these furnaces it was agreed that the value of mechanisation on small plants is largely a question of economics. It is possible to get as good an efficiency on a manually-operated plant as can be obtained by automatic control if the plant is well operated, but in the absence of regular, skilled attention the automatic control can give appreciable advantages in economy. (This question is raised again in Section 6.)

A type of furnace for burning a wide range of solid fuels in appliances of this size was described. This had a sloping grate, gravity-fed from the top. The fuel bed was kept thin and the region between the point of entry of the coal onto the grate and the combustion zone was kept very short to reduce swelling and agglomeration and so prevent blockage. There was a strong secondary air jet which was directed downwards across the front of the fuel bed and which helped to create a suction over the bed. The gases from the furnace chamber were removed by an air ejector, it being a feature of the design that air leakage through the bunker, with a consequent risk of combustion, was eliminated. A further paper on the control of this type of plant was concerned, not with the steady automatic operation of central-heating boilers, but with the regulation of the heat output in order to give a constant degree of comfort and maximum economy, and to ensure safety. Various intermittent and continuous methods of control were discussed though without engineering or operational details.

The final paper in this section dealt with the use of coal-fired heating appliances such as the modern open fire, closed stoves, cooking stoves and central-heating plant in Great Britain. The effect of the Clean Air Act and our present fuel supply position on the appliances used in this country were discussed and the possible advantages of the B.C.U.R.A. small-bore central-heating system mentioned.

(5) Furnaces

Two papers were presented on the coal firing of kilns for the manufacture of earthenware and pottery. It was stressed that here the advantages of fuel oil and electricity over coal were causing a fall in the use of solid fuel even though the newer methods improved the efficiency of the kiln and reduced the labour required. The moving-fire system for operating kilns for the manufacture of earthenware was described.

(6) Control, Testing and Boiler Availability

The final part of the Conference was concerned with various miscellaneous subjects, starting with a discussion on the new French Government order on the instrumentation of boilers. The need for adequate instrumentation was stressed but the possibility of getting a more satisfactory state of conditions by Government decree was viewed with a certain amount of scepticism. In the first instance a system of periodic examination had been instituted and this had been followed by suggesting—in some cases compelling—consultation with the Government by industry on matters concerning fuel utilisation. Later decrees make it a statutory requirement to have some method for controlling boiler efficiency. The difficulties of standardisation were stressed and it was also pointed out that instruments were of no use without efficient firemen. Automatic control would alter this position to some extent, but it was stressed that it was necessary to use a psychological approach to the firemen to get the maximum advantage from a plant. A case was quoted where, in a competition between firemen (all of whom were thought to be well trained and efficient), the best obtained an efficiency 60 per cent greater from a specific plant than the worst. What the difference would have been if bad firemen had been included was conjectural. A draught gauge was the first instrument to be insisted upon, followed by a CO_2 meter.

The fully-automatic control of large furnaces was described. It was mentioned that the measurement of coal feed rate was one important difficulty still to be overcome. Methods of superheater control were discussed in considerable detail, French practice being to use tilting burners or the recirculation of exhaust gas. Among miscellaneous points raised on this paper were: (*a*) the influence of the addition of water to steam (in desuperheaters) on water-side deposit formation; (*b*) the difficulty in measuring the oxygen content of flue gases, even with improved analysis instruments, owing to stratification; and (*c*) the correct variation in excess air with load. It was stated that in one case there was a six-minute lag between a change in steam temperature at the entry to a superheater and its detection at the exit. It was agreed that the more normal method of control in which the boiler follows the turbine was to be preferred to the reverse procedure, initially tried on the Benson boiler.

A further paper described the routine "direct" and "indirect" methods of determining the thermal efficiency of domestic-heating appliances. It was agreed that it was best to measure efficiency by both means as a check one against the other. The application of these techniques to large boilers was considered.

The last two papers were concerned with deposits and corrosion in boilers, the first being a survey of the influence of sulphur trioxide on low-temperature corrosion. The effects of the combustion temperature, inhibitors and mitigators (such as smokes), etc., on SO_3 formation were outlined, a dewpoint meter was described and various means of reducing the metal temperatures of the rotary air heater were listed. In the last paper, the corrosion of various types of steels subjected to the influence of flue gases at high temperatures was discussed.

(7) Conclusion

It is interesting to compare this Conference with the one in London a week earlier on pulverised coal, organised by the Institute of Fuel, and with other recent Conferences in this country. Obviously the scopes of these Conferences were different, though sometimes there was a surprising similarity between the discussions. For example, there was an air of familiarity about the discussion in Paris on the relative merits of wet and dry bottom p.f. furnaces. The discussion on automatic control of small boilers and the need for trained firemen to justify instrumentation followed closely some of the comments

made at the recent Harrogate Conference organised by the Combustion Engineering Association on the Annual report of the N.I.F.E.S.

Probably the most marked difference in approach was in the attitude to low-grade fuels. The continental countries take what we would consider to be very poor fuels as a matter of course and the design and installation of novel plant to deal with ash contents up to 75 per cent, V.M. (dry, ash-free) varying from below 10 per cent to approaching 80 per cent, and covering all size ranges for both grate and p.f. firing is well established.

There is little doubt that, while our problems and approach will be tempered to the particular requirements of our own fuel and economic positions, there is much to be gained on both sides from a free exchange of ideas across the Channel.

LIST OF PAPERS PRESENTED AT THE CONFERENCE
(IN ORDER OF DISCUSSION)

(1.01) R. A. Huchet, "Washing and Preparation of Coals."

(1.02) M. J. Moinard, "Future Trends in Coal Supplies."

(1.03) M. P. Bertrand (in collaboration with MM. Rabet and Mas), "Requirements of Manufacturers and Users of Plant."

(2.01) P. Dumoutet, "Application of Laboratory Tests to the Selection of Heating Coals (Domestic Appliances)."

(2.02) A. Hossard and G. Quesnel, "Determination of the Friability (Grindability) of a Fuel and the Fusibility of Ash."

(3.01) E. Wicke, "Kinetics of the Combustion of Graphitic Carbon."

(3.02) P. O. Rosin, "Mechanism of the Combustion of Pulverised Coal."

(3.03) R. Loison and G. Tissandier, "Experimental Study of the Combustion of Pulverised Coal."

(3.04) J. Boehm, "The Temperature of Radiant Wall Screens in Pulverised-Fuel Furnaces."

(3.04-A) J. Boehm and M. H. Viereckl, "Contribution to the Calculation of Mean Furnace Temperatures."

(4.01) P. Aubathier, "Hydraulic Transport of Sludge at the 'Emile Huchet' Power Station."

(4.02) J. Danze and L. Delvaux, "Drying of Coal for Use as Pulverised Fuel in Power Stations. Some Belgian Problems."

(4.03) J. Commissaire, "Transport of Pulverised Fuel by Air."

(4.04) M. G. Quesnel, "Problems Posed by the Increase in Mill Size."

(4.05) M. Robert, "Development of Mills for Pulverised-fuel-fired Power Station Boilers."

(4.06) M. Fournier, "Dry Bottom Pulverised-fuel-fired Boilers."

(4.07) G. Bouttes and A. Vellard, "Problems Posed by the Use of High-Ash Coals in Power Stations."

(4.07-B) H. Lenkewitz, "The Technique of Burning Lignite in Western Germany."

(4.08) W. Lenz, "Slag-Tap Furnaces."

(4.09) M. Krug, "Principles and Construction of the Vertical Cyclone Furnace."

(4.10) Mm. Seidl and Rosahl, "Horizontal Cyclone Furnace."

(4.11) Dr. Ing. Gumz, "Development of Pulverised-coal-fired Boilers in Germany."

(4.26) M. A. Folliot, "Transmission of Heat by the Flames and Flue Gas in a Rotary Cement Kiln."

(4.12) M. L. Juif, "Travelling-Grate Stoker with Pneumatic Injection of Fuel."

(4.13) M. Svoboda, "The 'Ignifluid' Grate."

(4.14) M. L. Stouff, "Stokers with Suspended Combustion (Stouff Process)."

(4.15) M. Lemaire and M. Henry, "The 'Lemaire' Grate."

(4.16) R. Vandeuelde and C. Gamet: "The Use of Bituminous Shale as Pulverised Fuel."

(4.17) M. A. Ramond, "The Utilisation of Waste Wood Products."

(4.18) P. Olmi and P. Sulzer: "The Combustion of Domestic Refuse."

(4.19) M. Besson, "Preparation and Combustion of 'Landes' Lignite."

(4.20) D. V. Velitchkovitch, "The Requirements and Possibilities of the Use of Peat in Heat Generation."

(4.21) M. Bernard, "The Use of Low-Volatile Coals and Large-sized Coke in Central Heating."

(4.22) P. Bertrand, "Recent Automatic Coal-Burning Appliances."

(4.23) M. Yribarren, "Study of a Central Heating Unit to Use Various Fuels."

(4.24) E. Bodmer, "The Control of Heat Release in Central Heating Units."

(4.25) J. S. Williams and W. C. Moss, "Coal-fired Heating Appliances in Great Britain."

(4.27) J. Massieye, "Evolution of Heat in Solid-fuel-fired Pottery Kilns."

(4.28) V. Bodin, "The Utilisation of Coal Fines in Earthenware Manufacture."

(5.01) M. F. Michel, "Application of the New (Government) Order on the Instrumentation of Boilers."

(5.02) P. Senechaut and R. Chaussard, "Automatic Control of Large Furnaces."

(5.03) G. Burnay; "The Laboratory Testing of Domestic Heating Appliances."

(5.03-A) G. Burnay, "The Testing of Large Boilers."

(5.04) N. Nedelko and G. Conanon, "The Dewpoints of Flue Gases from Solid-fuel-fired Appliances."

(5.05) M. J. Moreau, "The Problems of Corrosion by Hot Combustion Gases."

Standard Samples of High-Temperature Alloys

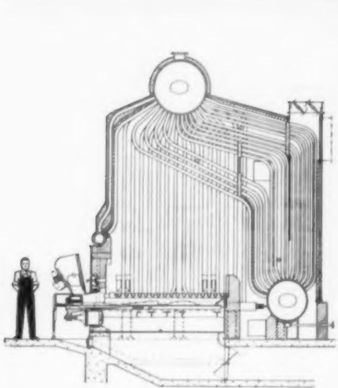
THE National Bureau of Standards is conducting a program on high-temperature standard samples at the request of the Navy Bureau of Aeronautics and the Wright Air Development Center. Standards for both spectrometric and chemical analyses are being prepared to cover the composition ranges of 17 commercial types of high-temperature alloys. Research on new and improved methods of analysis is also under way, to ensure accurate analysis of the standards and optimum conditions for their application in other laboratories.

The Bureau now distributes over 550 standard samples of chemicals, metals, and other materials to industrial and research laboratories for use in instrument calibration and process control. However, recent advances in jet aircraft, rockets, and guided missiles have brought about a need for additional standard samples to aid in controlling the composition of alloys that can withstand the high temperatures encountered in these applications.

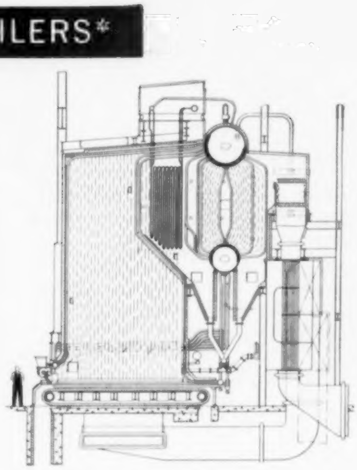
Such alloys are composed of heavy elements in a variety of compositions. The principal alloying elements are iron, nickel, and cobalt in various proportions, to which may be added lesser amounts of carbon, manganese, silicon, chromium, molybdenum, tungsten, niobium, titanium, and aluminum. In addition to the major alloying elements, the minor or trace elements present in the alloys, such as boron or zirconium, may have marked effects on their properties.

Because of the complexity of the alloys, chemical analyses are difficult and time consuming. On the other hand, optical and X-ray spectrometric methods offer considerable promise for rapid analysis provided that suitable standard samples of known composition are available for calibration of the spectrometers. The spectrometric standards now being prepared should meet this need.

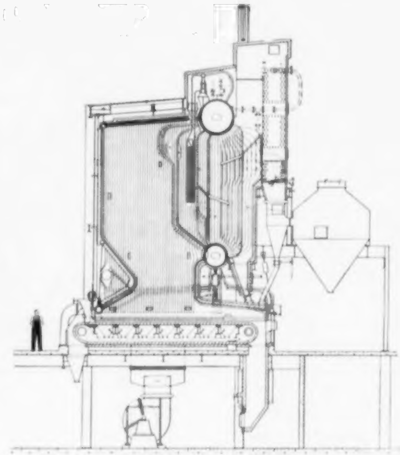
STOKER-FIRED BOILERS*



C-E Vertical-Unit Boiler, Type VU-10 — fired by a C-E Underfeed Stoker, Type E — VU-10 Boilers are available for capacities from 10,000 to 60,000 lb of steam per hr with pressures to 475 psi; superheat to 200 F in larger sizes. Also can be equipped with C-E Spreader Stokers, dump grate type.



C-E Vertical-Unit Boiler, Type VU-40 — fired by C-E Spreader Stoker, continuous discharge type — A baffleless boiler with capacities ranging up to about 300,000 lb of steam per hr, with pressures to 1,200 psi; temperatures to 950 F

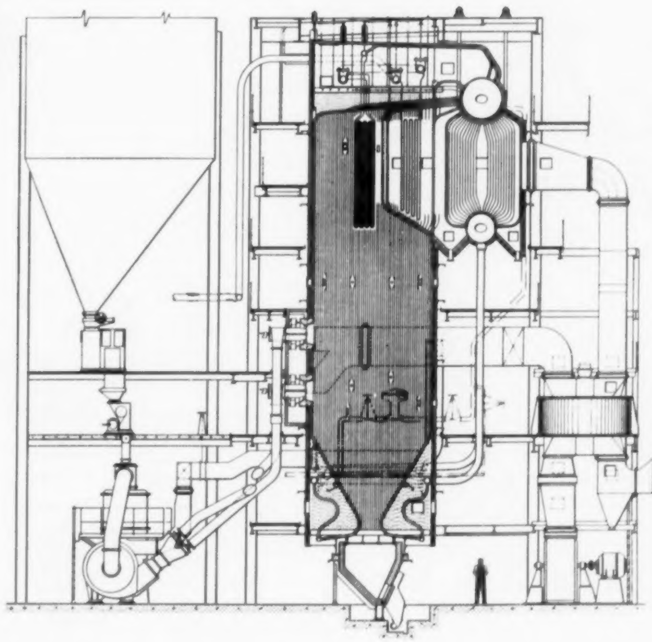


C-E Vertical-Unit Boiler, Type VU-50 — fired by C-E Traveling Grate Stoker — Units of this design are suitable for capacities up to about 150,000 lb of steam per hr; pressures to 1,200 psi; temperatures to 950 F.

BOILERS

equipment of firing

PULVERIZED COAL FIRED BOILERS*



C-E Vertical-Unit Boiler, Type VU-40 — using C-E Raymond Bowl Mills and Horizontal Burners — Capacities up to 600,000 lb of steam per hr, pressures to 1,200 psi, temperatures to 950 F.

These drawings are a few examples of the many units available for coal firing: all are readily adaptable to oil or gas firing.

COMBUSTION ENGINEERING

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Canada: Combustion Engineering-Superheater Ltd.

C-139

PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE

Abstracts from the Technical Press—Abroad and Domestic

(Drawn from the Monthly Technical Bulletin, International Combustion, Ltd., London, W.C. 1)

Fuels: Sources, Properties and Preparation

The Suitability of Coals from U.S.A. for Boilers with Slagging Furnace. J. Zorn. *BWK* 1958, **10** (May), 229–30 (In German).

The suitability was deduced from data on ash fusion temperatures given in two publications of the U.S.B.M. The study suggests that ash containing less than 75 per cent of SiO_2 + Al_2O_3 will permit trouble-free operation. For coals with ash of excessive fusion temperatures, admixtures of coal with ash of low fusion temperature or of lime or marl in amounts of 1 to 2 per cent of the weight of coal have proved useful.

Total Gasification of Coal. H. R. Hoy and D. M. Wilkins. *B.C.U.R.A. Monthly Bull.* 1958, **22,** Pt. 1 (Feb./Mar.), 57–110.

The development of total gasification of coal is reviewed under the headings: (1) Uses of synthesis gas; (2) Gasification processes; (3) Comparison of processes; (4) Gasification under pressure; (5) Cost of oxygen; (6) Impurities in the make gas; (7) Future developments.

Mechanical Handling

Pipeline Slurry Worry Nearly Over at Eastlake. Anon. *Elect. World* 1958, **149** (May 5), 46–7.

The difficulties encountered in the hydraulic transport of coal from the mine to the Eastlake power station are discussed and some of the modifications recently introduced are described. These refer to the sizing of the coal before its mixing with water and the removal of the water at the receiving end. It is believed that all difficulties have now been overcome and commercial operation of the pipeline can start soon.

Steam Generation and Power Production

Thermal Stresses Caused by Heating Thick-walled Hollow Cylinders. W. Endres. *Brown Boveri Rev.* 1958, **45** (Jan.), 21–8.

The mathematical analysis of the problem shows that thick walled pipes heated or cooled internally are subjected to greater thermal stresses than those heated externally. Heating produces high compression values which are generally not very dangerous, but repeated cooling produces tensile stresses which, together with stresses due to internal pressure, may lead to failure.

Overstrain and Bursting Strength of Thick-walled Cylinders. S. M. Jorgensen. *Trans. A.S.M.E.* 1958, **80** (Apr.), 561–70.

Stress distribution in elastically strained and partially overstrained thick-walled cylinders is described and a method of determining the most advantageous degree of overstrain presented. An empirical formula for bursting strength is proposed.

From *author's abstract.*

The Visual Qualitative Approach to Duct Design for Power Plants. M. J. Archbold. *Combustion* 1958, **29** (Apr.), 34–40.

An apparatus is described for investigating the flow in two-dimensional models of ducts, air preheaters and dust collectors. The work has been especially concerned with the installation of baffle or vanes to eliminate turbulence and improve laminar flow. Some of the results obtained are illustrated. The studies have led to considerable improvements in actual plants.

New Equation for the Evaluation of Tests on the Natural Circulation of

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W. H. Armacost Selected for ASME Medal Award

The American Society of Mechanical Engineers Medal, given for distinguished service in engineering and science, has been awarded to Wilbur Hering Armacost. The presentation will be made in December at the ASME Annual Meeting. Mr. Armacost is vice president-consultant and chairman of the technical committee at Combustion Engineering, Inc., New York.

Mr. Armacost's citation reads: "For his thorough knowledge of engineering, for his application of scientific principles and results of experimental research to design problems, for his outstanding leadership in the advancement of design and materials in the field of steam power generation, for his administration of engineering design, operation and research." Mr. Armacost is awarded the ASME Medal. It is one of the highest honors the Society bestows.

A pioneer in the development of design and materials adaptable to high temperatures and pressures, Mr. Armacost has been awarded over 75 patents. As vice president in charge of engineering at Combustion Engineering, Inc., he is credited with fostering such developments as the controlled circulation boiler and the supercritical boiler. The controlled circulation boiler, now used extensively by the power industry, incorporates many advanced design features developed by him.

Mr. Armacost began his career in 1916 after he graduated from the Armour (now Illinois) Institute of Technology with a Mechanical Engineering degree. He became a research engineer for Armour and Company. Later he went to the Ford Motor Company as a design engineer, a position he later held with the Superheater Company. Superheater also appointed him chief engineer.

Mr. Armacost joined Combustion Engineering in 1937 as chief engineer, becoming vice president of several of the company's divisions in 1943 and vice president in charge of engineering in 1948.

Water in Steam Generators. A. Zinzen and F. Schubert. *BWK* 1958, **10** (May), 216–8 (In German).

A new equation has been developed to take into account the observation that bubbles rising in the water carry with them considerable amounts of water. The equation has made possible a qualitative agreement of the results of tests by various authors.

Steam Generators (in Germany). O. Rosahl. *BWK* 1958, **10** (Apr.), 161–3 (In German).

Steam generators in large power stations must be able to operate efficiently at partial loads and to be started and stopped frequently. In this respect natural circulation boilers have an advantage over forced-flow boilers. For reasons of stability of circulation, a stabilizing region with high pressure drop has been introduced in the water flow of forced flow boilers. The severe stresses caused by rapid temperature fluctuations occur more often during cooling than during heating cycles and rapid control of the steam temperature is therefore imperative. Austenitic steels are more sensitive to temperature fluctuations than ferritic steels and are therefore avoided as far as possible.

Brown coal fired boilers with an output of 900 klb/h are already in operation, but bituminous coal fired boilers for 100 MW are still under construction. A Benson boiler with a pressurized vertical cyclone furnace and additional blast furnace or coke oven gas firing in the upper part of the furnace and without i.d. fan is the first of this kind in Germany. Tangent tube walls are becoming general in boiler construction. Forced-flow boilers are so designed that they are supported and not dependent.

Operational Experience with a Brown Coal Fired Cyclone Boiler. W. Jauer. *Mitt. V.G.B.* No. 53, 1958 (Apr.), 121–31 (In German).

The boiler is rated at 200 klb/h at 1250 psi and 975 F and equipped with two horizontal cyclones, each with its own mill and vent gas cleaning system, to enable operation at loads of 50 klb/h during the night. The boiler was to use mainly two kinds of brown coal, but alternatively also a bituminous-brown coal mixture or bituminous coal only. When fired with brown coal only, complete blockage of cyclone and slag screen was experienced because of the high CaO content of the coal; this could be alleviated by adding 5–6 per cent of bituminous coal. Fouling of superheater surfaces was caused by the use of raw brown coal and in the melting down of ash in the cyclone from two adjacent boilers fired with the same coal. Difficulties were further experienced by excessive flue gas temperatures, corrosion of the electrostatic precipitators for cleaning the vent gases and damage to the burner mouths. Acceptance tests gave a thermal efficiency of 90.6–90.3 per cent between 160 and 200 klb/h and a minimum load of 46 klb/h with liquid slag removal. After overcoming the initial difficulties, operating times of over 5000 hours are confidently expected.

Cyclone Fired Boiler With Blast Furnace Gas Furnace. J. Glöckner. *Mitt. V.G.B.* No. 53, 1958 (Apr.), 80–5 (In German).

The boiler is rated at 250 klb/h at 1850 psi and 985 F when fired with coal and at 175 klb/h when fired with blast furnace gas only and has two cyclones discharging the combustion gases into a secondary furnace which is separated by a slag screen from the tertiary furnace above; the blast furnace gas furnace is built on to the rear wall of the tertiary furnace. The

controls automatically start the cyclone furnace when gas flow or pressure are too low to generate the required steam volume. The experience with firing coal or gas alone and in combination has been very satisfactory; fouling was so low that the soot blowers in the tertiary furnace could be removed and deposits on the convection surfaces could easily be dealt with by the steel shot cleaner. Boiler efficiency with one cyclone in operation was 92 to 93.4 per cent with two cyclones 93.67 per cent and with blast furnace gas firing 88.6 to 89.2 per cent.

Recent Experience in Acid Cleaning of Boilers. H. Dorsam and R. Kretzer. *Mitt. V.G.B.* No. 53, 1958 (Apr.), 92–8 (In German).

Cleaning of new boilers (400 klb/h) with phosphoric acid did not prove satisfactory because the protective layer deposited on the tubes was soon dissolved by water and steam. Two further and larger boilers (900 klb/h) were, therefore, cleaned with inhibited hydrochloric acid by a specialist firm. This proved entirely satisfactory as shown by a comparison of the SiO_2 and Fe contents of water and steam during the first months of operation of the boilers. The procedure adopted for the two kinds of acid cleaning are set out in detail.

Acid Cleaning of New High Pressure Boilers. K. Rummel. *Mitt. V.G.B.* No. 53, 1958 (Apr.), 98–101 (In German).

The procedure employed at Fortuna power station for the acid cleaning of two 500 klb/h and two 720 klb/h natural circulation and two 900 klb/h Benson boilers is described. Inhibited hydrochloric acid at room temperature and without recirculation has been used and the various parts were cleaned in succession, the superheater tube bundles before installation and welding. Examination of tubes after cleaning and experience of boiler operation have indicated that all the mill and welding scale was effectively removed and deposits on turbine blades did not occur.

Solid Fuel Firing

Burning Coal in Pulverized Form. U. B. Yeager. *Pwr Engng* 1958, **62** (May), 69–71, 98, 100, 102.

The factors influencing power consumption in pulverizers and combustion conditions in the furnace are reviewed, especially moisture content, grindability, ash content, heat content, particle size, feeding and treatment of coal.

Liquid and Gaseous Fuel Firing

Oil Firing. H. Mika. *BWK* 1958, **10** (Apr.), 166–9 (In German).

A critical review of recent publications is presented, dealing with: (1) Merits of various types of oil burners; the Sulzer pressure atomization burner with adjustable air exit velocity is singled out as one of the best modern designs; (2) boiler fouling and corrosion; (3) application of and experience with dolomite; (4) controls; (5) boilers specifically designed for oil firing; (6) air pollution.

Water-Side Corrosion and Water Treatment

Feedwater Specifications for the Power Stations of the Central Electricity Authority. R. Ll. Rees and F. J. R. Taylor. *Mitt. V.G.B.* No. 53, 1958 (Apr.), 86–92 (In German).

An account is given of the feedwater and make-up water treatment plants in C.E.A. power stations and the specifications laid down for the permissible contents of residual oxygen, carbon dioxide, ammonia, iron and copper, calcium and magnesium and oil, the electric conductivity, the pH value, and the addition of sulfite. The reasons for arriving at the specified values are discussed and methods for measuring the various quantities described. The specifications for boiler water are treated in a similar way.

The A.B.C.'s of Demineralizing. F. N. Kemmer. *Combustion* 1958, **29** (Apr.), 41–4.

A review of the basic principles, methods and applications of demineralization.

Corrosion and the Destination of Corrosion Products in a High Pressure Power Plant. R. C. Tucker. *Corrosion* 1958, **14** (May), 19–22.

Observations made over several years of corrosion in the feedwater-steam-condensate circuit of a high pressure boiler-turbine plant, the origin of corrosion and concentration of corrosion products in certain parts of the circuit are described. The application of morpholine to reduce corrosion of condensate lines and of hydrazine or sodium sulfate as dissolved oxygen scavenger are discussed. Corrosion is promoted by insufficient cleaning of new boilers, by low load operation and by periodic shutdown during week ends if no provision is made to deaerate the feedwater during such periods, since a high

concentration of dissolved oxygen is then available to attack all metal surfaces.

Gas-Side Corrosion and Deposits

Accelerated High Temperature Oxidation Due to Vanadium Pentoxide. K. Sachs. *Metallurgia* 1958, **57** (May), 224–32.

The third part of the review deals with possible remedies suggested by various authors.

Flue Gas, Ash and Dust

Measurements on Brown Coal Electrostatic Precipitators. P. Wiemer. *Mitt. V.G.B.* No. 53, 1958 (Apr.), 102–3 (In German).

The guaranteed separation efficiency demanded of electrostatic precipitators has been raised from 90 per cent in 1953 to 94–96.5 per cent in 1956–1957. The investigation has shown that some of the precipitators were unable to fulfil the guarantees, while others exceeded them, but it was impossible to discern the responsible factors. The separation efficiency of electrostatic precipitators has lately been greatly improved by increasing the electrical output of the rectifier plant and higher secondary voltage (50–55 kv), especially the latter.

Better Boiler Efficiencies Are Expected with New Precipitator. J. J. Trainor. *Elect. World* 1958, **149** (Apr. 21), 60–1.

The precipitator is to be installed between the cyclone separator and the air preheater and designed to clean 652,000 cfm of flue gas at a temperature of 690 F in 8000 tubes of 3 in. diam at a gas velocity of 32 fps. The tubes will be cleaned for two minutes by a suction cleaner creating a momentary gas velocity of 200 fps. The two collectors will together remove 98.5 per cent of the fly ash in the gas. Because of the absence of fly ash particles, the nuclei required for condensing SO_3 and H_2O are missing and the formation of H_2SO_4 in the air preheater is prevented; it can, therefore, be operated at lower gas exit temperatures.

Power Generation and Power Plant

Heat and Power Generation in Public and Industrial Heat Power Stations. Part III: Public Heat-Power Stations. T. Geissler. *Energie* 1958, **10** (Apr.), 137–44 (In German).

The merits and demerits of public heat and power stations, the factors influencing the efficiency and economics of such plants, the selection of steam parameters and heat distribution by steam and hot water are discussed.

American Power Conference in Review. I. Anon. *Combustion* 1958, **29** (Apr.), 51–4.

Abstracts are given of papers presented at the 20th American Power Conference under the headings: (1) Steam Turbines; (2) Deaeration.

Britain's Largest Power Concentration. Opening of C Station Brings Capacity of Hams Hall Site to 930 Mw. Anon. *Elect. Times* 1958, **133** (May 15), 787–9.

Hams Hall A with a capacity of 249 Mw was completed in 1939, Hams Hail B with 321 Mw in 1949 and Hams Hall C with 360 Mw was officially opened at the beginning of May 1958. The C station contains six boilers, each rated at 550 klb/h at 950 psi and 925 F, and six 60 Mw turbogenerators. Each boiler is provided with tangentially firing tilting burners supplied from three mills (plus one reserve) and when burning coal with C.V. of 8500 to 9000 Btu/lb, 20 per cent ash and 15 per cent moisture, the thermal efficiency is 86 per cent and the overall thermal efficiency of the station 29.3 per cent.

Experience With Cyclone Furnace. Anon. *Elect. Times* 1958, **133** (May 1), 718.

In a paper to the I.M.E., R. McCulloch, J. R. Willets and A. Richardson gave details of the experience during the first year of operation of the horizontal cyclone fired boiler installed at the Kynoch works of I.C.I. Ltd. Tests showed the thermal efficiency of the boiler to be 88.1 per cent, and even higher efficiencies may be possible with some coals. Difficulties arose with coals where the slag had a high viscosity and with operating the cyclone at low partial loads. The modifications introduced during the year now permit operation down to 40 per cent of full load. Particle size of the pulverized fuel appeared unimportant and dust emission did not exceed 0.033 grain/cu ft at the precipitator exit. Altogether the operation of the boiler was regarded as very satisfactory.

A New 600 C High Pressure Power Station in the Reisholz Works of Feldmühle A.G. F. Wehrberger. *Mitt. V.G.B.* No. 53, 1958 (Apr.), 69–80 (In German).

Calculations showed that the paper mills could be supplied most economically with 80 klb of process steam and 20 Mw of electric power from a plant generating steam at 2400 psi and 1130 F, the steam to be expanded in a 31.5 Mw topping turbogenerator to 590 psi (the pressure of the old boiler plant) and the steam not required for process to be passed through a 12.5 Mw condensing turbo-

generator. The two Benson boilers are each rated at 110 klb/h and contain a lower slagging furnace with four corner burners, an upper furnace reduced in cross section by 36 per cent and separated from the lower furnace by a slag screen. The slag hole is near the rear wall and the lower part of this is parabolic, to concentrate the temperature on to the hole; additionally, hot furnace gas may be recirculated and drawn up the slag hole. The upper furnace walls are all part of the superheater; there is also a pendant tube platen superheater in the upper part of this furnace and a final austenitic superheater is in the convection pass which contains also some of the evaporating tube surfaces and the economizer. The burners are designed for firing a mixture of brown coal (delivered pulverized) and bituminous coal (pulverized and stored in intermediary bunkers), oil (up to 30 per cent load) or gas and fly ash (stored in bunkers). The two mills per boiler have each an output of 6.9 t/h and are supplied with hot flue gas from the upper furnace and with cold flue gas from the chimney, so that the whole mill and intermediary bunker system are operating in an inert gas atmosphere. The raw coal bunker is lined with special concrete, the intermediary bunkers with "Saekaphen" and steam heated. The ferritic and austenitic steels used for the boilers, pipelines and turbine are tabulated. Also described are: (1) The turbines; (2) the pipelines; (3) feedwater treatment; (4) controls; (5) coal and ash handling; (6) electrical plant.

Materials and Manufacturing Processes

Metallizing Superheater Tubes. E. Wicklund. *Pwr. Engng.* 1958, **62** (Apr.), 103.

To replace loss of material on superheater tubes due to erosion by fly ash, a metallizing process was successfully applied. The entire furnace was cleaned in a conventional manner and the affected parts then sandblasted twice; metallizing was applied immediately after the second sandblast to prevent oxidation of the clean surface. For the same reason the operator wore clean canvas gloves so that his hands should not come into contact with the metal. A thickness of 0.01 in. was applied all around the tubes to prevent flaking, but later experience showed this to be unnecessary. The cost was $2300 against $25,000 for replacing the affected superheated loops in two boilers, and the time required was only a fraction of that for complete replacement.

Heat Transfer Conference Meets in Chicago

Better than 500 engineers met at the Second National Heat Transfer Conference jointly sponsored by the ASME and the American Institute of Chemical Engineers. The program for the typical power engineers could be classified as "long hair" since many of the subjects were treated from a highly theoretical approach. Yet, many of the subjects under discussion were of direct interest to power people. Below we abstract a few of the papers.

Heating Coils

"The Design of Heating Coils for Storage Tanks" by **David Stuhlbarg,** the Proctor & Gamble Co., covered the thermal design and some phases of physical design of heating coils for storage tanks. It presented methods for calculating heat losses, formulas for determining the coil area, and a discussion of various factors affecting

the coil layout and location within the tank. Experimental data were given supporting some of the material presented. Results may be applicable in part for other applications, such as process or cooling coils, though these were not specifically covered.

Once the amount of coil needed has been determined, the author reported, the other factors affecting coil design, such as size, location, and configuration must be resolved. Material selection for corrosion resistance is also important, but it is outside the scope of this paper.

Cost considerations usually determine the size pipe used. If the coils are shop fabricated, 2 in. and 2½ in. are reportedly the most economical sizes, with little difference in price when compared on a surface square foot basis; but if the coils are field fabricated, the reports we receive favor either 2 in. or 1½ in. As a result, and since it is rarely known in advance whether a coil is to be field or shop fabricated, we have standardized designs at 2 in., unless some other factor modifies the choice. For example, if water is the heating medium, it may be necessary to reduce the size in order to secure a satisfactory heat transfer coefficient.

Plate coils and fintube tank heaters are used at times in lieu of bare pipe and have some advantages. Plate coils are easy to clean, and fintube heaters offer considerable heat transfer area in a compact bundle that can be brought into a tank through a 20-in. manhead.

Tank suction heaters are also used occasionally instead of coils. Their function is somewhat different in that they heat only the stream being pumped off rather than the entire tank contents. They are applied particularly to very viscous liquids which are difficult to pump when cold. They are designed and built much like exchangers and may be of either shell and tube or fintube construction. They differ in that they are mounted through the wall of the tank and the end of the shell is left open to serve as a stock inlet.

The advantages of suction heaters are: they save on fuel costs since tanks are left cold, and there is no time required for reheating the tank before using the stock. The disadvantages are their applications are limited, and the tank cannot be pumped completely empty through the heater.

A number of problems on coil configuration and location have been investigated.

In one test it was discovered that, for stocks which partially freeze, complete coverage of the tank bottom

is important. A tank containing tallow, which normally melts completely around 100 F, was observed during reheating after three months storage in moderate weather. It was noted that by the time the tank reached 120 F, most of the tank wall had become warm to the touch, indicating the stock inside had remelted; but two areas, 3 to 4 feet wide and extending up from the ground 5 feet, remained cold for considerably longer. These were not completely warmed until the bulk of the stock had been seriously overheated. The tank was later drained and entered and it was found that the coil was of 2-in. pipe set on 24-in. centers well distributed across the bottom, except at the two areas which were last to be heated. These areas had a small amount of unmelted tallow left near the wall. There were also a few spots halfway between successive turns of the coil which retained a small amount of unmelted tallow. Subsequent discussions with factory personnel led to the belief that this behavior was typical for fat storage tanks. To alleviate the condition, our standard design for this type of material now calls for an encircling coil ring within 6 in. of the tank wall, and sufficient bottom coverage so that no point on the bottom is more than 12 in. from the coil.

G. T. Atkins and **N. O. Felps**, Humble Oil & Refining Co., presented a paper entitled "The Heat Exchanger—An Economic Study." In this report, maintenance experience with a large group of refinery heat exchangers during a two-year period has been tabulated for the purpose of forecasting maintenance expense. A hypothetical example has been worked out to illustrate a method of using this information.

During the two-year period, 1955–1956, 1500 shell-and-tube exchangers in service at a large refinery were studied. Some were relatively new and had not required maintenance. Others had seen nearly their full life. Most were installed 10 to 15 years ago. They covered a great variety of size, design, and type, representing products of 25 manufacturers.

For purposes of this study, the following were excluded: tank-heating coils; boxcoolers; engine-jacket water and lube-oil coolers; air-conditioning items. The greater number considered were in the size range of 1000 to 3000 sq ft of surface, with removable bundle, floating-head construction. All were in accordance with the recognized codes for good materials, design, and construction.

The raw data need interpretation. It will be apparent that an adjustment is called for on the basis of the

statistics, namely, the number of exchangers in service. As refining and petrochemical processes have improved in technology and grown in capacity and complexity, there has been a substantial increase in the number of heat exchangers in service. Adjustment is made by using divisors, increasing from about 500 to 600 for the oldest group up to the 1500 in service during the two-year period of June 30, 1955 to June 30, 1957.

During the two-year period, a total of 45 shells was retired. An average life of 25 years is found when one uses a divisor of 500 to 600. That is: in a refinery which is maintained at constant capacity, 20 to 25 new exchangers installed each year and lasting an average of 25 years would correspond to a total of 500 to 600 in service.

Experience shows that 25 years is indeed a typical life of a refinery-type exchanger. But, obviously, if corrosion is severe or the materials not entirely suitable for withstanding corrosion, then that exchanger would be expected to have a correspondingly shorter life. Also, obsolescence is a factor, for when exchangers are retired they are not, in our experience, replaced in kind. Better materials, designs, and processes are available and are used in the modern units which take the place of the old ones.

After other prime considerations have been satisfied, the relative cost factors—as expressed in the concept of the economic balance—then control the design and sizing of process heat exchangers. Cleaning and repairs are expensive, but on the other hand the first cost of the installation can become excessive from a too conservative selection of the excess heat transfer surface or of the materials specified. The goal in mind is the minimum overall cost when considering the opposing factors weighted one against the other.

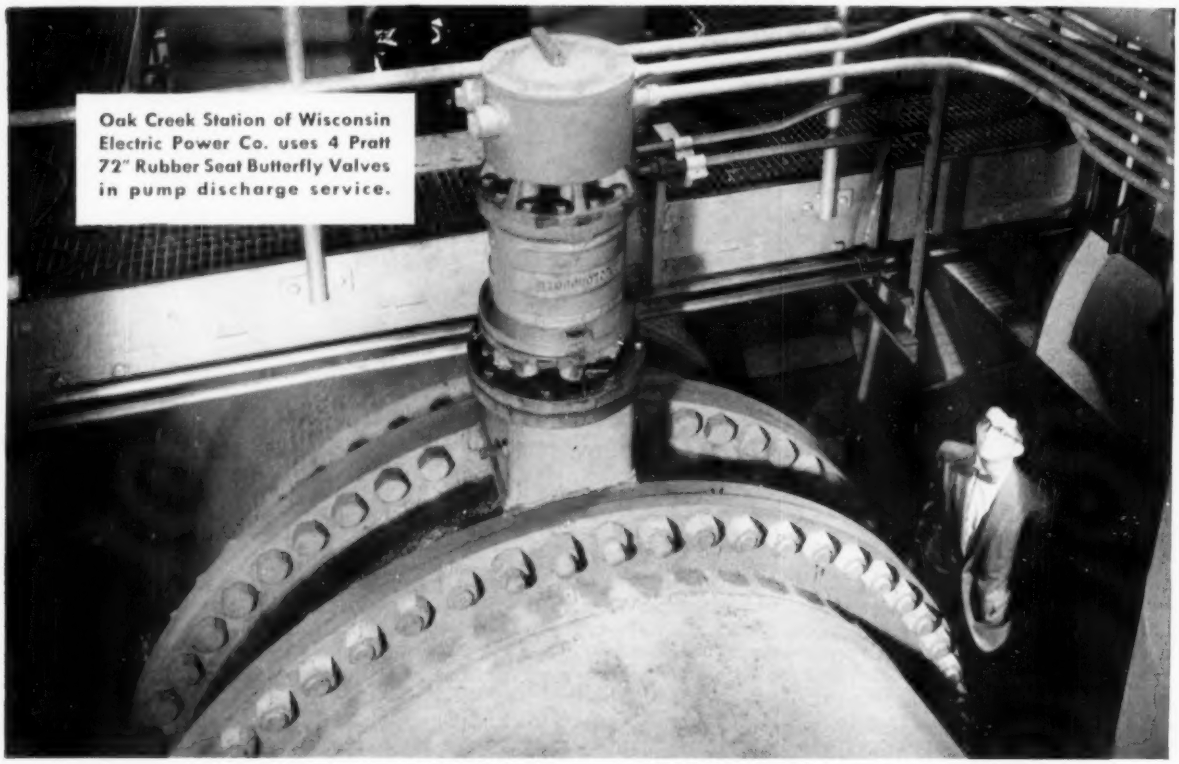
"The Pressure Drop of Condensing Steam in Horizontal Pipes" was presented by **R. J. Dunn** and **D. Stuhlbarg,** Proctor and Gamble Co. This paper presents a new equation for computing the pressure drop of condensing steam in horizontal pipes and experimental data supporting it.

Steam flow measurements were made by a Brown Flow controller installed as a meter. Pressure readings were taken near the meter orifice and at the coil ends using dial pressure gages with 2 psi markings. These instruments were all checked for accuracy before installation.

Initially, pressures were read with the throttle valve open and the tank agitator off. The agitator was then turned on and the pressure read at various settings. After taking readings on the first coil, the instruments were shifted to the second tank and similar readings taken.

Test results were given and compared with pressure drops calculated. A friction factor of 0.012 as reported by Bottomley was used in the calculations. This figure, incidentally, was confirmed by Benjamin and Miller who reported factors of 0.0116 to 0.0131.

Pressure drops were calculated for coils by equations for the test flow rates and compared with the observed pressure drops. One equation gave figures which were high for each and every flow rate, with an average deviation of +62.6 per cent. The second equation gave figures both high and low with an average deviation of +8.5 per cent and with 16 out of 25 of the calculated pressure drops within ±10 per cent of the measured figures.



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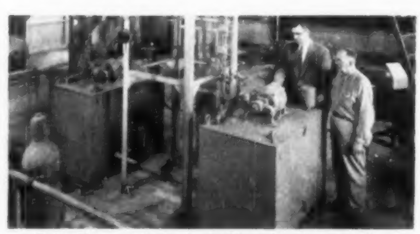
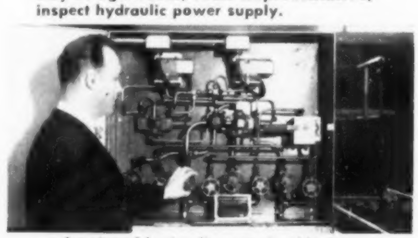


Felix Kazmerchak, Ass't. Plant Engineer, and Ray F. Egebrecht, Pratt Representative, inspect hydraulic power supply.



Interior of hydraulic control cabinet.

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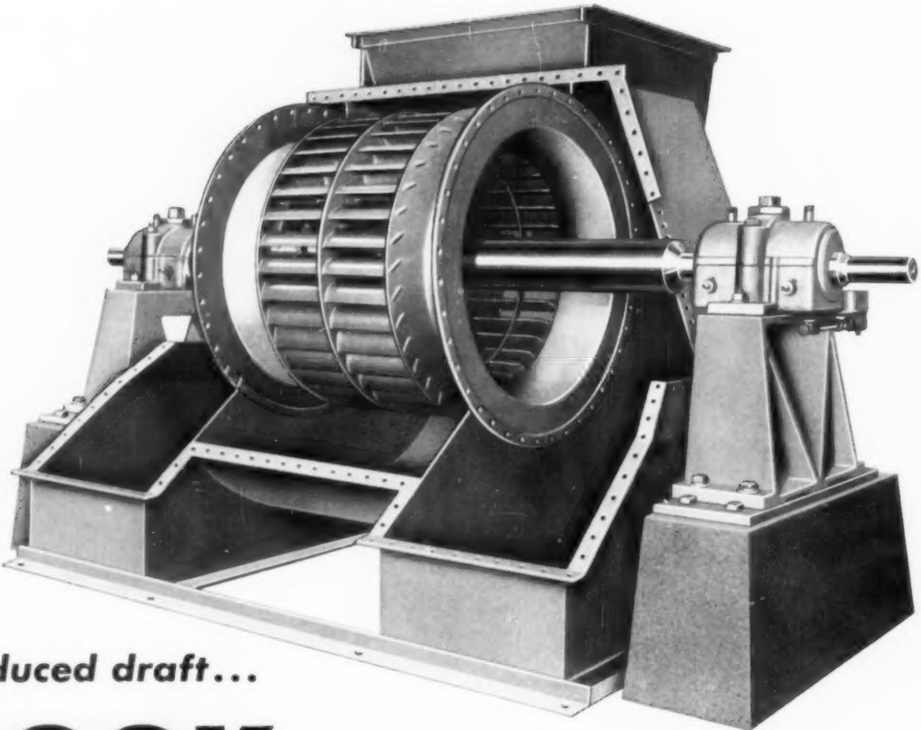

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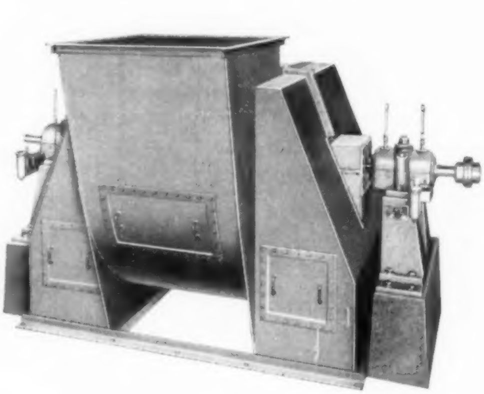

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